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DESIGN STUDY OF A KINEMATIC STIRLING ENGINE FOR DISPERSED SOLAR ELECTRIC POWER SYSTEMS

UNITED STIRLING (SWEDEN)

1980
FINAL REPORT

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for

U.S. DEPARTMENT OF ENERGY
Solar Thermal Energy Systems

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16. Abstract

Objectives for the study are to demonstrate future kinematic Stirling engine design, performance and production cost.

The concept evaluation shows that the four cylinder double acting U-type Stirling engine with annular regenerators is the most suitable engine type for the 15kW solar application with respect to design, performance and cost.

Results show that near term performance for a metallic Stirling engine is 42% efficiency. Further improved components show an impact on efficiency of the future metallic engine to 45%. Increase of heater temperature, through the introduction of ceramic components, contribute the greatest amount to achieve high efficiency goals. Future ceramic Stirling engines for solar applications show an efficiency of around 50%.

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Appendix 1 — Stirling engine heat pipe systems
Appendix 2 — Discussions of beyond 1985
Appendix 3 — Optical systems of the advanced dish — Stirling module
Appendix 4 — Special heat flux distribution calculations for a P40 engine

REFERENCES

1 INTRODUCTION

This final report contains the results of a study contract awarded by the NASA Lewis Research Center to develop a conceptual design of a nominal 15 kW electric solar Stirling engine. Conceptual designs were evaluated and developed for both a free-piston and kinematic-type Stirling engine for small dispersed solar powered applications in identical parallel path programs.
This volume contains only the results for the kinematic Stirling engine study which was performed by United Stirling under subcontract to MTI. A separate volume contains similar information for the free piston Stirling engine.

The study performed configuration definition studies, including a detailed parametric evaluation of the selected concepts, with a final ranking of all attractive configurations. These included single and multiple cylinder engine with rhombic and crank shaft drive, different seal systems and heat exchanger arrangements. For the conceptual design a 4-cylinder double crank shaft engine with annular regenerators, with the power level increased to 20 kWe output from the alternator, was selected.

The study also addressed the interface of the engine heater head with the solar reciever and collector. Cases for solar radiation directly impinging on the heater tubes, and condensing sodium heat transfer via heat pipe transport were considered.

The final report describes the study processes and the results leading the way to the detail design of an efficient Stirling engine generator with concentrated solar radiation heat input.

1.1.1 Background

As part of the Solar Thermal Program in the Division of Solar Technology U S Department of Energy, studies and experiments are being conducted for Central Power Systems application and Dispersed Power Systems Applications. In support of these systems, the Advanced Technology Branch of the above division is conducting research on advanced systems and subsystems for increased efficiency and decreased cost of electricity from solar energy input. One of these systems is referred to as a point-focusing, parabolic dish collector-receiver. This type of solar collector-receiver, when coupled with a heat engine and generator set, can provide electrical energy for low power applications (kilowatt range). When coupled together in a much larger array they can produce power in the megawatt range. The NASA through the Jet Propulsion Laboratory (JPL) and the Lewis Research Center (LeRC) has the responsibility to conduct research and develop the concept for the Division of Solar Technology of DOE. The LeRC has the responsibility of developing the engines and power conversion equipment for the point-focus systems.

This study evaluates the concept definition of kinematic-type Stirling engines for single parabolic dish applications with a nominal output of 15-20 kWe. The Stirling engine systems offers the potential for high solar energy to electrical power conversion efficiencies and therefore reduced collector size resulting in lower total system costs.

1.1.2 Background of United Stirling

United Stirling is a research and development company, owned by the Swedish Government and FFV. The company's main goal is to continue the development of the Stirling engine and to adapt it for commercial production - to realize its potential as a reliable, economical and environmentally acceptable means of converting energy.

United Stirling was founded in 1968 as a licensee of N V Philips in Holland, a company that has been engaged since 1938 in research and development on the 150 year old Stirling Principle.

In 1970, United Stirling built the first in house developed Stirling Engine, based on the single acting Stirling principle; this engine had two pistons in each cylinder.

Two years later, United Stirling decided to concentrate development work exclusively on the double acting Stirling engine. It was becoming increasingly more evident that the Stirling engine was an attractive power plant for the vehicles of the future and in the application the double acting engine offers a number of advantages; it is simpler, more compact and less expensive, and it can be adapted to various types of drive systems.

The results that United Stirling have obtained for efficiency, power/weight ratio, starting time and response time for power control systems have been so satisfactory that we are now intensifying the efforts to adapt the Stirling engine for production.

In early 1977 the first prototype of a new configuration was tested, a 40 kW U-engine with parallel cylinders called P40. The engine configuration has been the basis of United Stirling's engine development as well as a reliable laboratory test bed engine.

The P40 design and experience from testing P40 engines is the base for development work on engines in the automotive Stirling engine development program, sponsored by DOE and NASA, in which United Stirling is participating.

1.2 Requirements

1.2.1 Introduction

The use of energy from the sun for other purposes than heating requires an efficient heat-engine. The energy from the engine will most suitably be converted into electricity to be practical. Use of the power from the sun at places where the insolation level is low, seems not justifiable with this type of application.

The solar collector cost is a major part of the total first cost for such a system, and that makes the engine cost less critical, but still an important consideration. On the other hand the engine efficiency is critical for keeping the collector dimensions small and consequently the overall system cost low.

1.2.2 Scope of Work

The subcontract specified work on the following tasks from United Stirling:

TASK I — KINEMATIC ENGINE CONFIGURATION DEFINITION STUDIES AND PARAMETRIC ANALYSIS

Two heat input concepts were to be considered for an advanced parabolic dish collector-receiver system for operation of a 15 kWe Stirling engine with an alternator. The first was a direct coupled solar receiver integrated with the engine; the second was an indirect receiver connected to the engine heater by a heat transport loop. Technology expected by 1985 was to be incorporated.

A parametric analysis was to be performed on the three best engine configurations to calculate efficiency, for heater temperature between 1200°F and 2000°F, at 4 power levels and 2 coolant temperatures.

The effect of solar insolation profile on engine design performance, and the selection of an optimum design point to maximize yearly kW hour production, was to be considered.

United Stirling was to rank order all concepts and provide technical and economic justification.

TASK II — KINEMATIC ENGINE IMPLEMENTATION ASSESSMENT

An assessment was to be made of the potential of the most attractive configuration, selected from Task I, for development to a production engine status and implementation in solar thermal power systems. The assessment was to include:

Scope

1) A qualitative evaluation of state-of-the-art of the key technology included in the conceptual design.

2) Evaluation of the potential of the conceptual design engine for manufacturing by mass production.

3) Identification of the key design features affecting engine durability and maintenance during the life cycles of 5, 10 and 20 years, on the solar duty cycle.

4) Evaluation of design concepts for power level growth potential in areas of engine and component efficiencies, and temperature - pressure - rpm limits.

TASK III — KINEMATIC ENGINE CONCEPTUAL DESIGN OF TEST BED ENGINE

1) A conceptual design, selected by NASA from Task I studies, of a test bed engine was to be prepared for fabrication and operation in the 1980 time period. A list of seven technical criteria was given for assessing the design choices.

2) An assessment of control requirements and modes based on 5 variations of heat input and 15 minute heat storage was to be made.

3) An overall energy balance for 4 power ranges was to be made.

4) An auxiliary heat source for engine testing independently of solar input was to be designed, and consideration of direct heating vs. heating from a remote receiver was to be made.

5) The design was to be fully defined by preparing design layouts and specifications, making thermal and stress calculations, and preparing control schematics.

TASK IV — KINEMATIC ENGINE SYSTEMS INTERFACE REQUIREMENTS

Coordination with MTI of engine heater head designs with receiver heat storage/collector concepts was to be made to resolve subsystem interfaces. Interface data were to be presented to MTI weekly, based on design concepts in Task III.

1.2.3 Design requirements

Design requirements initially stated in the contract were used for the configuration definition in Task I. For the conceptual design in Task III some changes were made. The design requirements and targets were as follows:

Power	Configuration definition	Conceptual design
	15 kWe 3 phase 60 Hz	20 kWe
Heater head temperature	1200 - 2000°F	1500°F
Cooler temperature	70°F and 150°F	110°F
Target efficiency	42% engine	40-45% engine
Receiver heat storage	15 minutes	None
Operation mode	Constant speed Application not specified	Constant speed Power to grid

Receiver/heater concepts for both direct and indirect (heat pipe) heating were studied using an insolation profile from Lancaster, California.

Based on these design requirements the object was to show the design and performance of a Stirling engine, which over its lifetime gives the minimum system cost per produced kWh.

The engine design reflects advanced principles and incorporates technology likely to be available in 1985.

1.3 Summary

1.3.1 Configuration study – Task I

All types of kinematic Stirling engines — both of single and double acting type — were surveyed for use in the solar application. For each engine component a number of critical factors were identified and evaluated. A parametric study was performed and a number of engines were conceptually designed. After initial screening, four engine concepts were studied in greater detail. These are shown in figures 1, 2, 3 and 4.

1) single acting, single cylinder rhombic engine
2) single acting two cylinder engine
3) double acting, four cylinder U-engine with annular regenerators
4) double acting, four cylinder U-engine with separate regenerators

Analysis showed no significant differences in performance for the four design concepts.

This left the choice of most attractive engine to be based on risk, development status, and scaleability to higher power levels.

The recommended advanced engine concept is the four-cylinder double acting U-engine with annular regenerators. This recommendation was accepted by the NASA project manager (see 2.9).

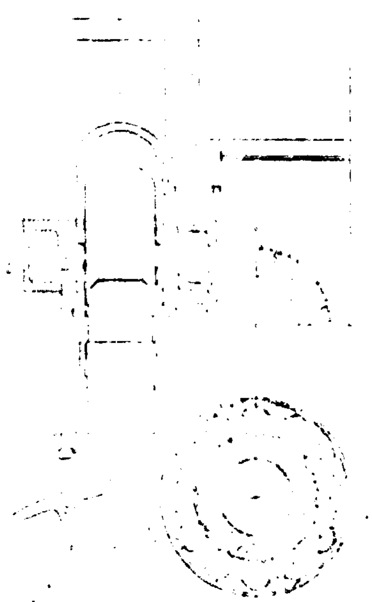

FIG 1: Single-acting, single cylinder engine

FIG 2: Single-acting, two cylinder engine

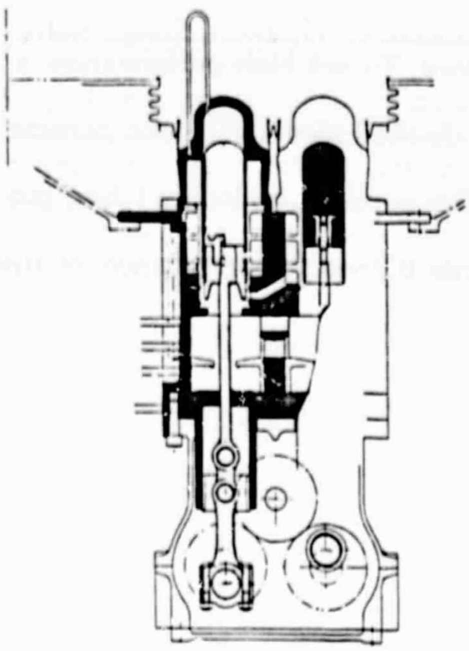

FIG 3: Double-acting, four cylinder engine, annular regenerators

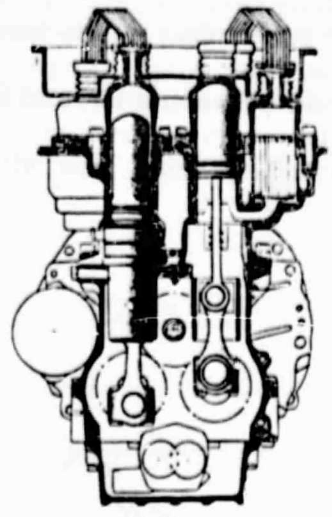

FIG 4: Double-acting, four cylinder engine, separate regenerators

1.3.2 Conceptual design – Task III

The four-cylinder double acting U-engine with annular regenerators has been analyzed and designed more in detail. An optimization of this engine concept has been performed to meet the assumption of 20 kW electrical output.

Heater designs for both indirect and direct insolation were produced. The heater for indirect insolation can have almost any shape (fig. 5). The Stirling engine cycle analysis will dimension the tube parameters - length, number, diameters.

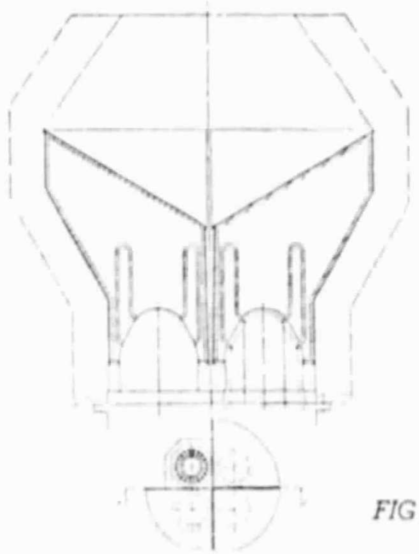

FIG 5: Heater for indirect heating, including heat-pipe

The heater for direct insolation has a number of additional design restrictions dictated by the solar heat reciever. To get high performance, a relatively even temperature distribution on the heater is required. Engine cycle parameters for the inner tube side shall meet outer tube parameters as tube length, heater cage diameter, heater cage cone angle, spacing between tubes and finning. This necessitates longer tubes, bigger dead volumes and unheated tube length compared to the optimal tube arrangement. This will of course influence performance of the engine. (Fig. 6)

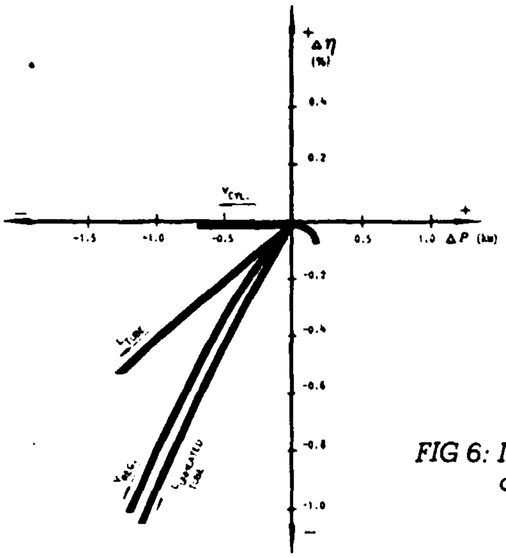

FIG 6: Influence on heater performance of different parameters

Three different heater designs were produced. (Fig. 7) Performance results show

- power variations within 2 kW
- efficiency variations within 2%-units

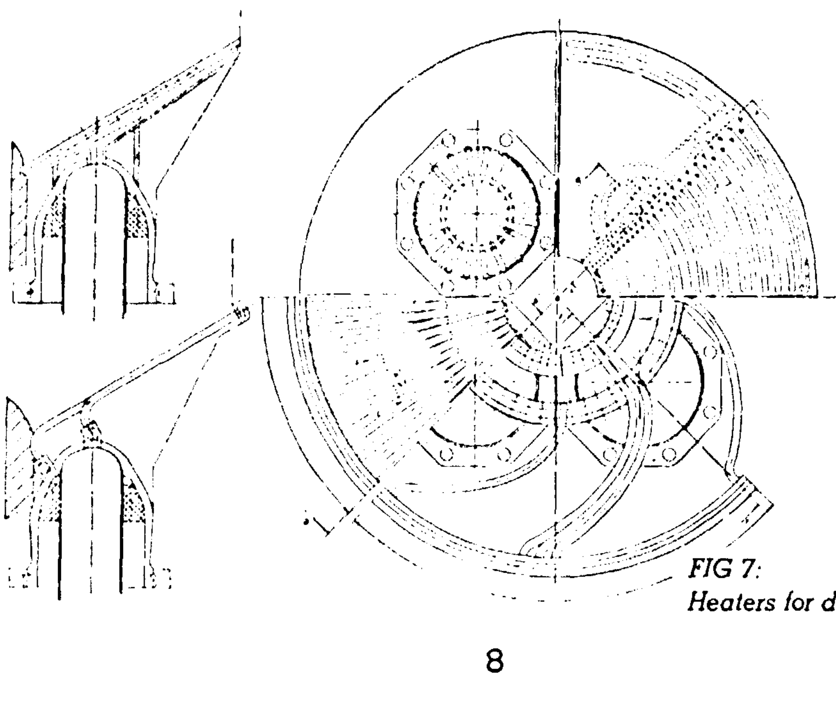

FIG 7: Heaters for direct heating

The arrangement of the cylinder/regenerator housings depend much on heater tube design and cold engine part design. In this study special emphasis has been placed on evaluating heater materials for use in solar application to meet a 50.000 hour life requirement.

In figure 8, only the heat pipe version was optimized. For the other three designs, only the heater designs were changed while the cylinders, regenerators and crank case parts remained the same as for the heat pipe design.

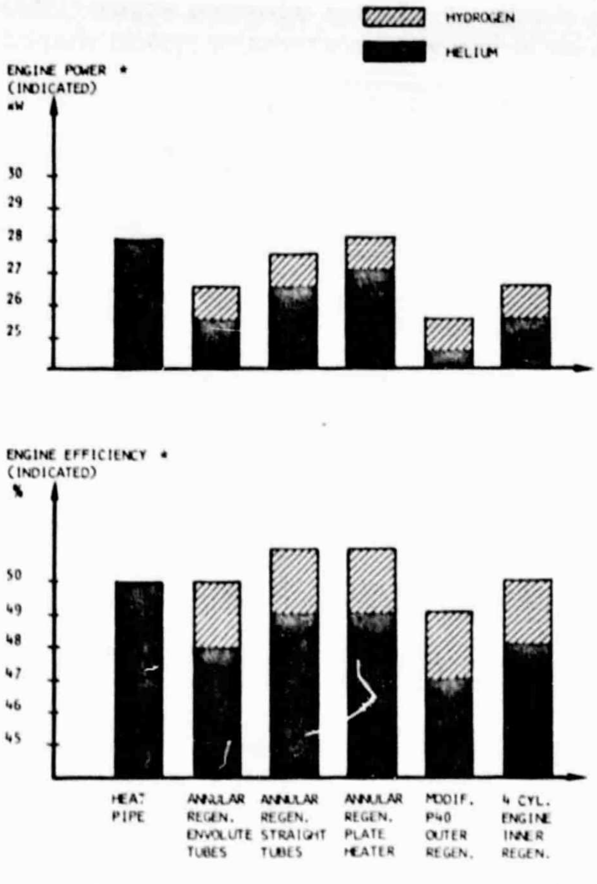

FIG 8: Engine performance for different heater designs

* *Cycle efficiency and power, based on no auxiliaries and no mechanical losses.*

The seal system, being one of the critical Stirling components, has been evaluated in detail. A recently developed modified sliding seal system, termed the PL seal, which is under successful testing in our laboratory, seems applicable for operation in inverted position in a solar application. Results from testing indicate that this seal system will meet the long life requirement for this application. As of January 3, 1980, over 39.000 hours of successful running has been accumulated on a total of 42 of the new PL seals. No failures have occurred, wear has been negligible and no oil has been detected in the working spaces of the engines.

The main dimensions for engine/alternator combination are (fig 9, 10)

- length 1000 mm
- width 325 mm
- height 620 mm

Estimated weight for the engine will be close to 200 kg (including all engine auxiliaries and subsystems).

Optional drive systems - for example swash plate or wobble plate - can be incorporated in the engine design for a further advanced engine (1990 engine), which will result in an in line engine/alternator system shaped like a cigar.

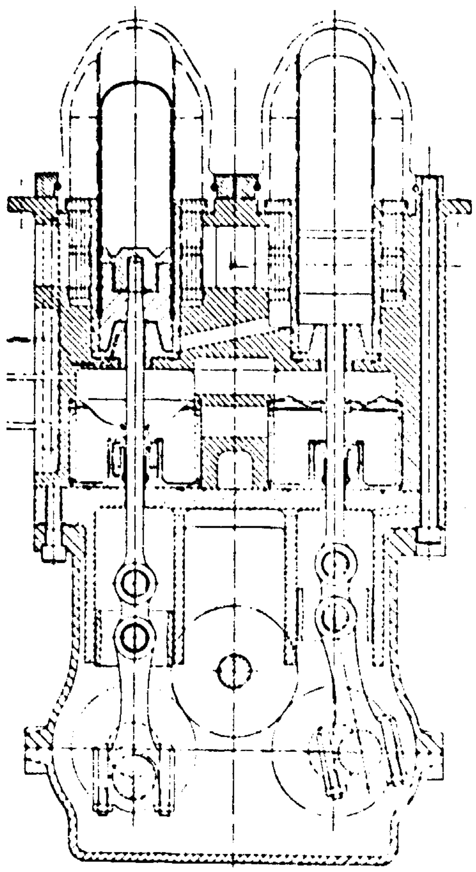

FIG 9: Cross-section, advanced engine

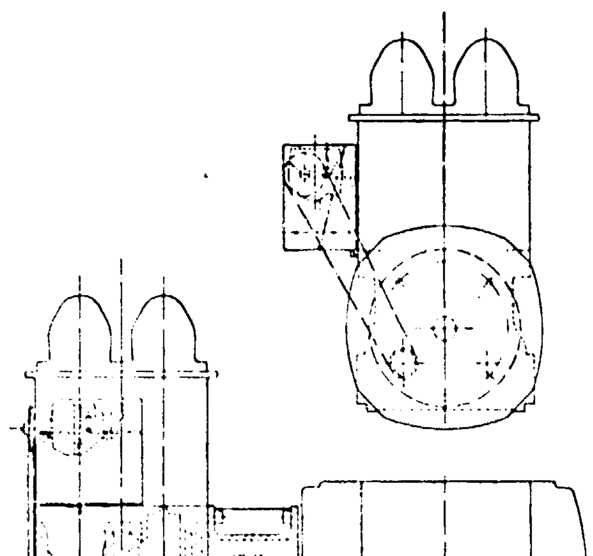

FIG 10: Engine/alternator design

1.3.3 Engine specification and operating conditions

The physical data of the recommended engine for indirect heating and operating conditions are:

Bore	54 mm
Stroke	43 mm
Pressure ratio (P_{max} / P_{min})	1.52
Number of cylinders	4
Swept volume per cylinder	99 cm^3
Number of heater tubes	12/cyl
Tube dimension	
(inner diameter/outer diameter)	4/6 mm
Tube length	258 mm
Number of cooler tubes	608/cyl
Tube dimension	
(inner diameter/outer diameter)	1/2 mm
Cooler length	65 mm
Regenerator type	Annular
Regenerator cross section area	4.860 mm^2/cyl
Regenerator length	70 mm
Filling factor	35%
Seal type	Pumping ring/diaphragm
Drive system type	U-engine two crankshafts
Alternator type	Induction (20 kWe output)
Input energy	54 kW
Losses — Rejected heat	26.5 kW
— Mechanical	5.5 kW
Engine efficiency*	41 %
Losses — auxiliaries	1 kW
Shaft power output (corr 20 kWe)	21 kW
Alternator losses (η 0.95)	1 kW
Engine and alternator combined efficiency	37 %
Output power — electrical	20 kWe
Engine speed	1.800 rpm
Working gas pressure	15 MPa
Working gas	Helium
Heater temperature	800° C
Coolant temperature	43° C

*Engine efficiency includes mechanical friction

When using indirect heating, helium must be used as working gas. If direct heating is assumed, hydrogen can be used as working gas from engine point of view. The performance increase when changing from helium to hydrogen (assumed is also change from indirect to direct heating) is in the order of 1 kW of power and 2%-units of efficiency, for a design optimized for helium. (Fig 11 and 12)

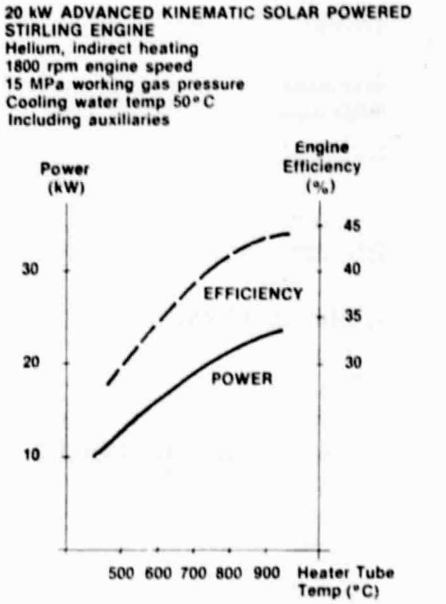

FIG 11: Performance, advanced engine

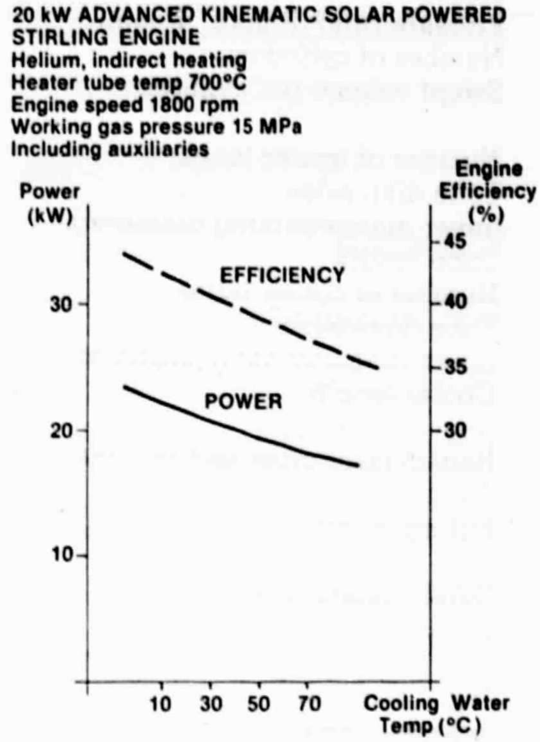

FIG 12: Performance, advanced engine

1.3.4 Controls (fig 13)

The main control principle of the Stirling engine is to keep the heater temperature constant. The output power variation will be controlled by the working gas mean pressure. In this application, only solar power input is assumed. The heater temperature will control the working gas mean pressure and output power from engine/alternator will depend on solar input power.

Start-up and shut-down of engine/alternator have been studied but require a more detailed analysis and further assumptions, before an operational sequence can be established.

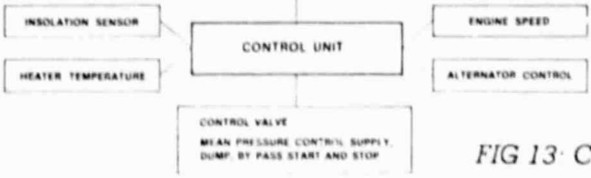

FIG 13: Control system

1.3.5 Implementation assessment – Task II

The Stirling engine of today demands further development for meeting the requirements of an advanced engine for solar application. Performance, cost and life of a number of components have to be improved to make the engine fit this application. The present component status, key technology and development required for the advanced engine have been evaluated. (Fig 14)

Component	Critical factors	Key technology	Technology Requirement
Heater head	Cost, life	Cast housings, brazed tubes	Improving present technology, new brazing techniques
Regenerator	Cost	Thin metal plates*	New manufacturing technology
Cooler	Cost	Aluminium dimpled tubes	New manufacture technology
Cylinder/piston Pistondome/seal	Life	Close tolerance seal or piston rings	New technology improved material or coating technique
Pistonrod/seal	Life	Sliding seal	Improving present technology
Cylinderblock/ drive	-	Cast block, U-drive	Improving present technology
Power control	Cost, life	Mean pressure control-sliding valve, or magnetic valves	Improving present technology
Control systems	Cost	Electronic system (micro computor)	Improving present technology

FIG 14

*"Thin metal plates" refers to an axially rolled sheet regenerator which is both perforated and dimpled for better flow distribution and for reduction of axial heat conduction.

Production cost for an advanced engine including auxiliaries and sub-system (but excluding alternator) is 1.200 $ at a production rate of 100 000 units per year. This evaluation is based on Swedish experience - comparison indicates a 10 - 20% lower production cost of engine in USA at current exchange rates. Cost for different production rates has been studied. (Fig 15)

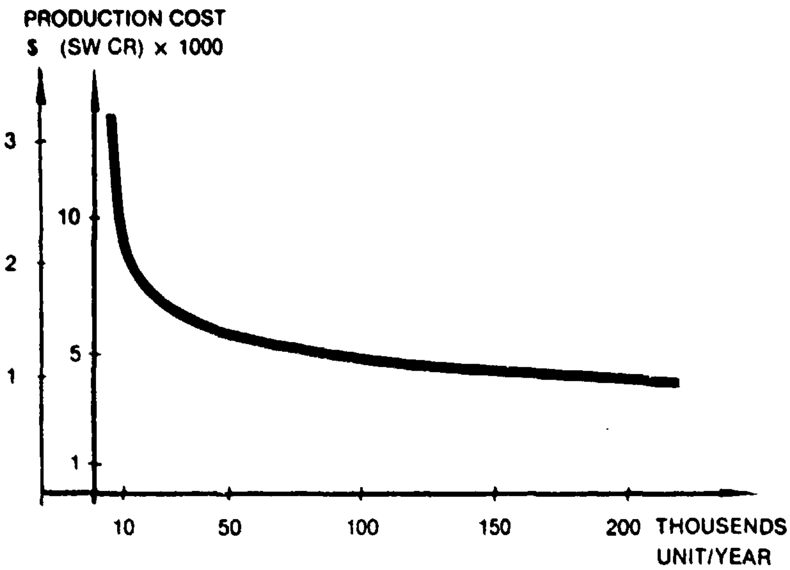

FIG 15: Production cost at various production volume

The cost evaluation shows the comparison of components which indicates the high heater cost due to use of expensive high temperature resistant materials. Also the relation between material (or prefabrication) and labour for the different components is a result from the production cost evaluation. (Fig 16)

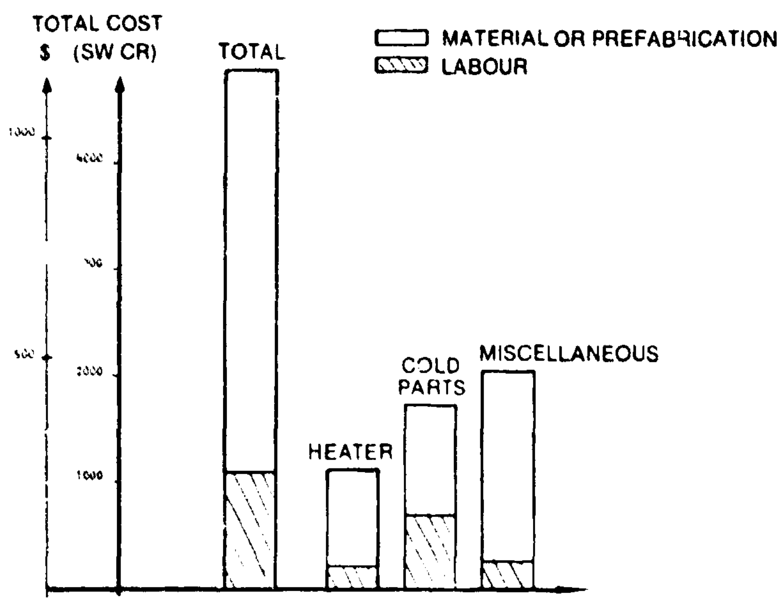

FIG 16: Component system cost — labour and material

The four-cylinder double acting concept with annular regenerators has been analyzed for scaling to higher power levels. The concept, including four housings containing both cylinder and regenerator is expected to have an upper power limit around 40 kW_{el}. The limitations are the increased housing dimensions - diameter and wall thickness. At higher power levels it is recommended to use separate pressure vessels for cylinder and regenerators. The increase in efficiency will be rather small when scaling up engines. Increased performance is mainly caused by smaller parasitic losses. (Fig 17)

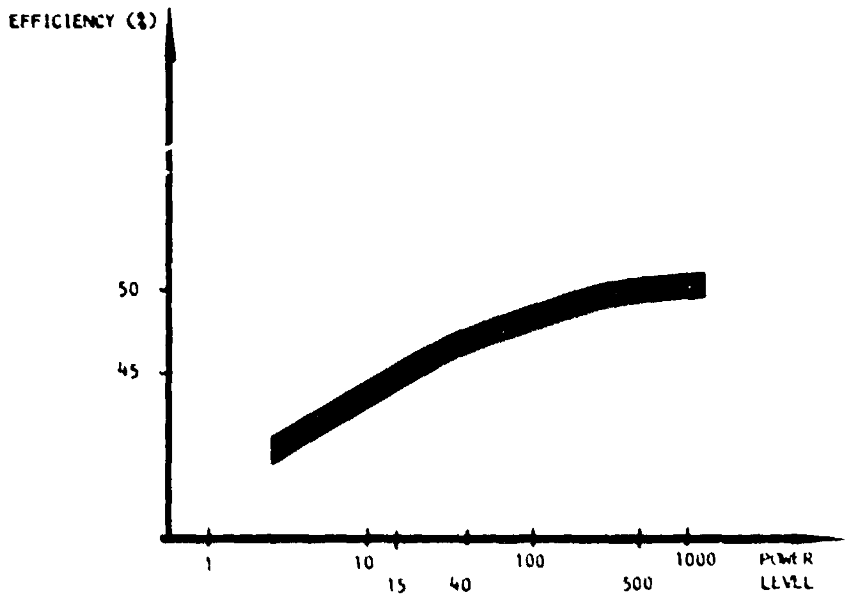

FIG 17: Engine efficiency at different design power levels

2. CONFIGURATION STUDY

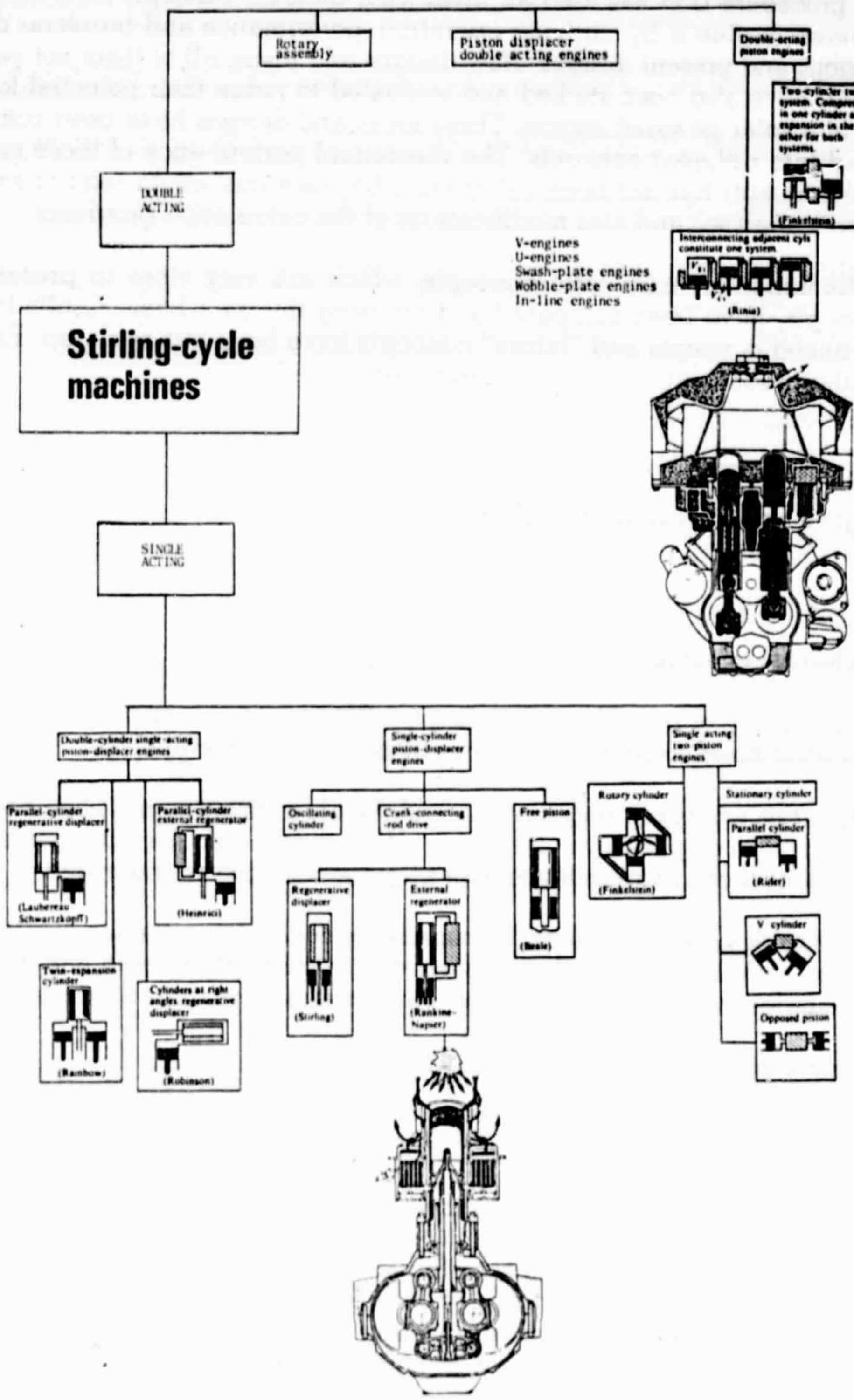

FIG 18: Stirling engines (After Walker, ref 3)

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2.1 Introduction

The procedure USS has used for identifying different concepts for a solar powered engine is by studying operation, performance and problems of previous and present designs. New designs and ideas, all of them not yet tested, have also been studied and evaluated to judge their potential for use in a solar powered engine. These ideas and designs have been combined into different concepts. The theoretical performance of these engine concepts has not been calculated, because this would require extensive analyses and also modifications of the calculation programs.

Performance for near term concepts, which are very close to present concepts, have been computed and by using this as a base, results for advanced concepts and "future" concepts have been extrapolated. Fig 18 shows schematically the concepts considered.

2.2 Engine and component evaluation

2.2.1 Technical overview of potential concepts

An inventory of possible concepts for a Stirling engine has been made and scrutinized to select the alternatives for use in the analysis.

Within this work both single acting and double acting engines have been studied, also engines with cylinders in line or in a circular arrangement as well as these engines types with all possible drive configurations.

In our work some "impossible" or unsuitable concepts for a solar powered engine have been rejected, as for example configurations with link drive and engines with cylinders in line. The remaining configurations are:

- single acting engines, rhombic, V or U drive
- double acting engines, cylinders in circular arrangement, U, V or swash-plate type

From previous experience of the different engine concepts as for example, inhouse development of single acting rhombic engine, double acting V and U engines, as well as knowledge about Ford/Philips swash-plate engine program, and experience with direct (normal combustion system) and indirect heating (design and testing of single acting, one cylinder engine and design of a double acting V4-engine, both for heat pipe operation) we have selected a number of concepts for continued studies, as follows:

1) Single acting engine
- one cylinder (rhombic drive)
- two cylinders (ordinary crank shaft)
- annular regenerator
- one or more separate regenerators
- indirect heating
- direct heating

2) Double acting engine
- four cylinders (U-drive)
- annular regenerators
- one or more separate regenerators
- indirect heating
- direct heating

2.2.2 Component/subsystem/principles and critical factors

The design of a Stirling engine includes a number of components/sub-system/principles, all of which must be taken into consideration, when evaluating different engine concepts.

Component/subsystem/principle	Alternative	Critical factors
Working priciple	Single/double	Losses, forces
Heating	Direct/indirect	Flux distribution Sodium
Heater head	Tube/Plate	Temperature distribution Stress Dead volume Flow losses
Cylinder	One/several	Dead volume Heat losses
Regenerator	Annular/Separate	Heat losses Dead volume
Seal system	Sliding/clearance	Gas leakage Oil leakage Wear Volume Friction
Drive system	U/V/Rhombic/ Swash-plate	Mechanical losses Torque variations
Control system	Temperature, pressure, amplitude, stroke, phase	Controlability Principle Volume

2.3 Single versus double acting engines

Both double- and single acting engines require a phase difference between the cold and hot volumes.

The single acting engine has a piston system where it is possible to have an optimized phase between the displacer piston and working piston. However, the phase difference is often limited due to the drive configuration and its working principle.

The double acting Stirling engine principle connects the upper side of one piston with the lower side of the following. This is continued one full turn. The minimum number of cylinders are 3. With respect to drive configuration 4 cylinders are most suitable. The double acting engine must have a fixed phase difference between the pistons (90° for 4 cyl).

The phase difference between the hot and cold volumes of a single acting engine depends on the motion of the power piston and the displacer piston (see fig 19). The ratio between cold and hot volume will also depend on drive system. This ratio must not be too low which can occur if the drive system lay-out is incorrect.

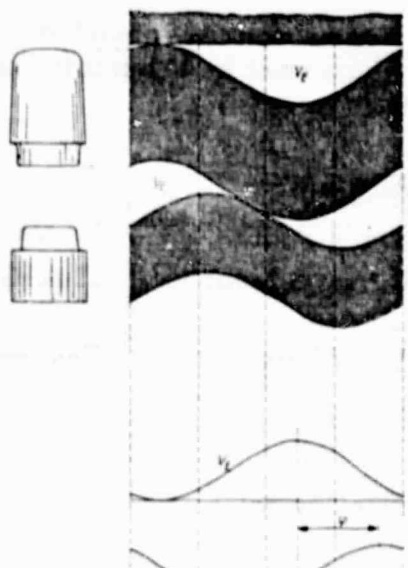


FIG 19: Motion of pistons in a single-acting engine

Typical values for drive system phase difference between pistons are 60° which will result in a volume phase difference of 115 - 120° and a volume ratio of about 1. (Fig 20)

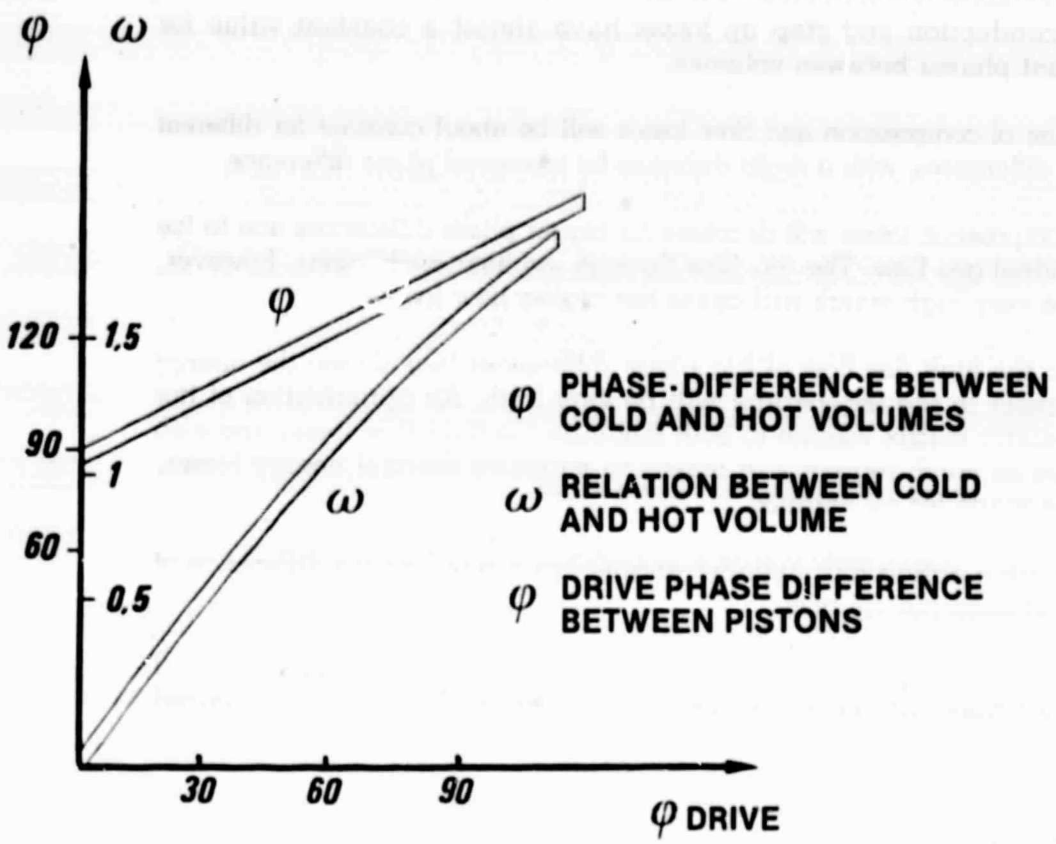


FIG 20: Drive system comparison in a single-acting engine

2.3.1 Losses

A number of losses included in the Stirling cycle vary depending on the value of this phase difference:

- Compression losses - losses due to adiabatic compression of the gas and due to the phase difference between the volume and pressure variation which cause a deviation from the ideal cycle.
- Flow losses - losses due to gas flow through the heat exchangers and the ducts.
- Conduction losses - losses due to conduction through heat exchanger walls and ducts from hot side to cold side.
- Step up losses - losses due to piston motion (gas between cylinder wall and piston dome).
- Regenerator energy flow losses - losses due to gas flow through regenerators without heat transfer.
- 2nd harmonic losses - losses due to the design of the drive mechanism.

Heat conduction and step up losses have almost a constant value for different phases between volumes.

The sum of compression and flow losses will be about constant for different phase differences, with a slight decrease for increased phase difference.

The compression losses will decrease for bigger phase differences due to the more ideal gas flow. The gas flow through the heat exchangers, however, will be very high which will cause the higher flow losses.

Due to the high gas flow at big phase differences (see above) the energy flow losses in the regenerator will be very high. An optimization of the regenerator matrix volume to both minimize the fluid flow losses and also to have as much regenerator matrix to minimize thermal energy losses, will be made for all engines.

The double acting four cylinder engine has a fixed phase difference of 90° between the volumes.

The single acting rhombic engine with displacer and working piston will have a phase difference between the volumes of 115°- 120° (optimized value and limitations due to the design of the drive system).

A comparison between the double acting and single acting engines will give the result:

- no significant difference in the cycle performance
- the design of the different engines will influence performance

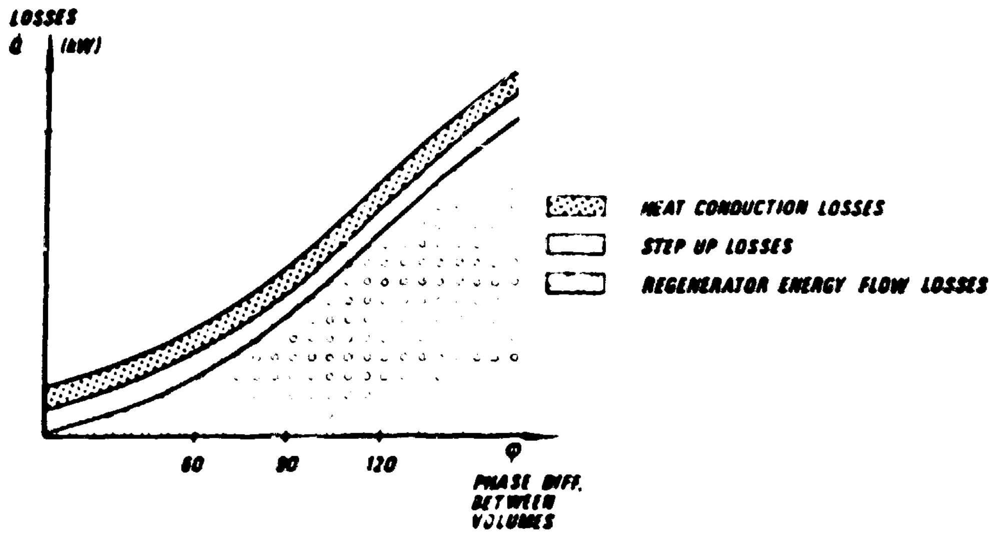


FIG 21: Losses at various phase differences

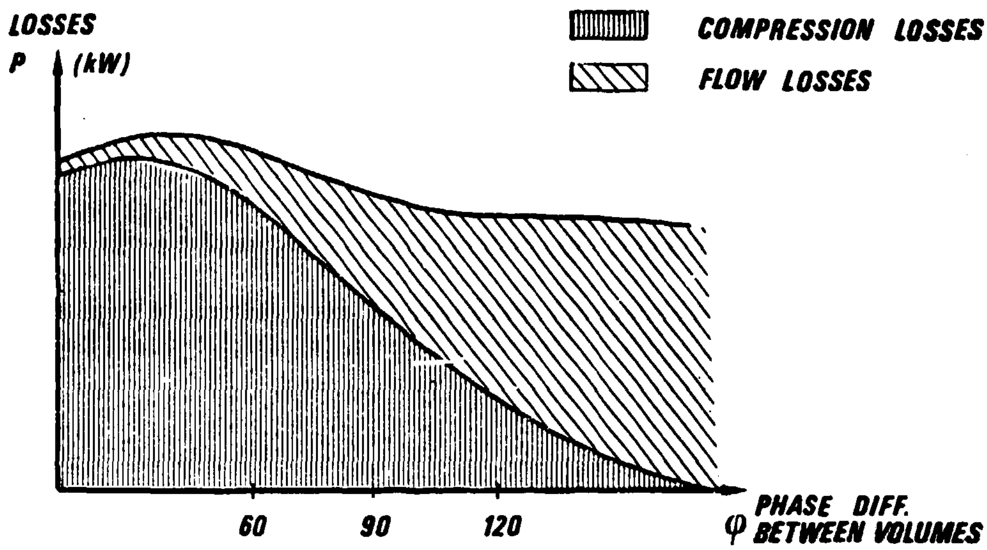


FIG 22: Losses at various phase differences

2.4 Engine analysis

2.4.1 Heating system

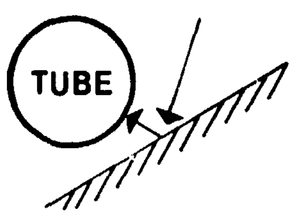
Both direct and indirect heating can be used for the Stirling engine. The design principle of the engines - the hot parts - will differ which will give differences in performance.

2.4.1.1 Direct heating

The Stirling engine can be heated directly by solar insolation. An uneven heat flux will occur if the directly heated surface geometry is not matched to the flux distribution from the parabolic mirror and/or internal reradiation are not accounted for in the design process.

An uneven insolation distribution gives a spread of temperatures in the heater and reduces the maximum possible efficiency.

An uneven temperature distribution around the heater tubes will always exist unless part of the insolation is redirected to the back side of the reciever tubes, as shown in the following sketch.



Absorption of all insolation from the collector requires lengthening of the tubes forming the walls of the cavity.

The Stirling cycle efficiency will then be decreased due to excessive dead volumes in the heater. A detailed explanation is given in section 2.4.5.

One advantage with the directly heated concept is that it permits the use of hydrogen as working gas, which will give increased performance.

With direct heating, thermal storage becomes more difficult to implement. The heat capacity storage in the metal of a heater unit for direct heating is very limited.

2.4.1.2 Indirect heating

With indirect heating no solar radiation falls directly on the engine heater tubes. Instead the heat is transported from a heated surface to the heater tubes by a liquid metal - usually sodium. With the use of indirect heating temperature maldistribution in the heater head of the engine is avoided. However, the liquid metal heat pipe and its interface to the engine require development.

The working gas must be helium (since hydrogen diffusion through the heater tubes is unacceptable) which will decrease performance. On the other hand the temperature variation will be very low ($\pm$ 5°C compared to $\pm$ 25°C for fossil fuel heating) and a higher mean working temperature can be used resulting in increased performance.

2.4.2 Heater head

A heater head for direct insolation can be designed with either independent tubes or milled channels in a plate, providing the strength is sufficient for the design temperature and pressure. As stated above the main problem is to cover adequately the whole receiver surface with working gas passages without excessively increasing the gas volume. By using an intermediate high conductivity matrix - such as a copper plate - this is more easily accomplished and creates a more even temperature distribution. Independent tubes - on the other hand - represent design simplicity.

If indirect heating is used the design has very few limitations. The heater tubes may have any shape. The heat transfer from sodium in the heat pipe is very high and the heat transfer inside the tubes on the gas side will govern the tube dimensions.

Relative performance of engines, with different heater heads is shown in fig 23.

Note that hydrogen as working fluid provides better performance than helium for direct heating, but a helium heat pipe Stirling is better at high speed. At medium speed the performance difference is less noticable.

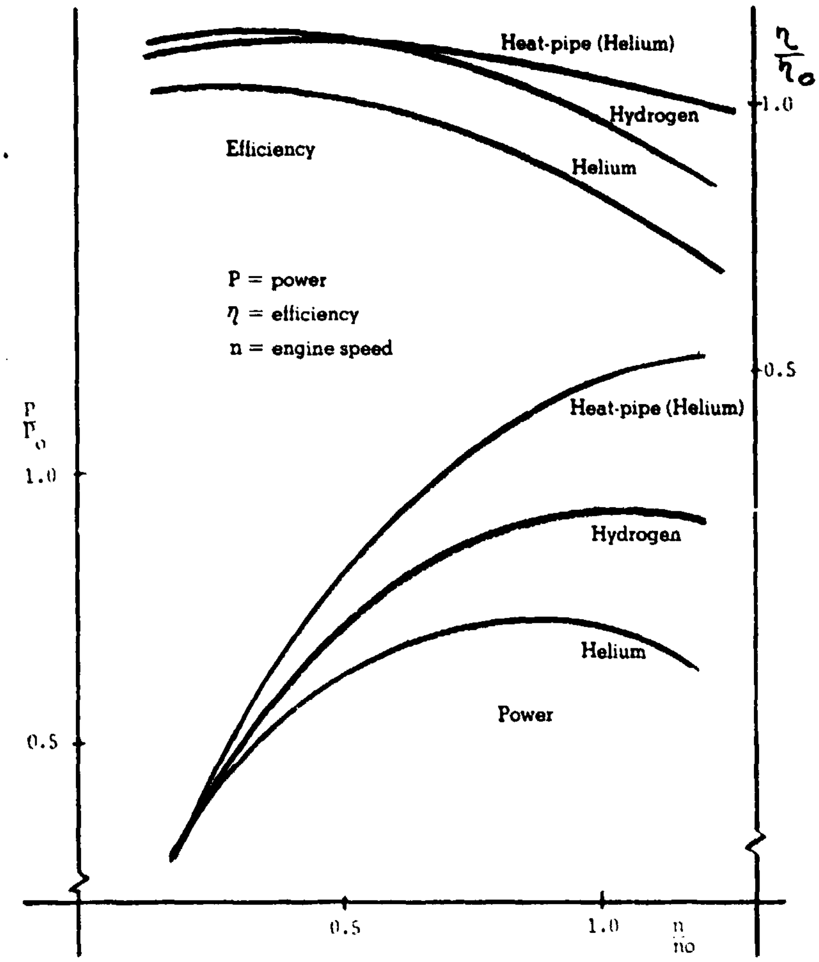


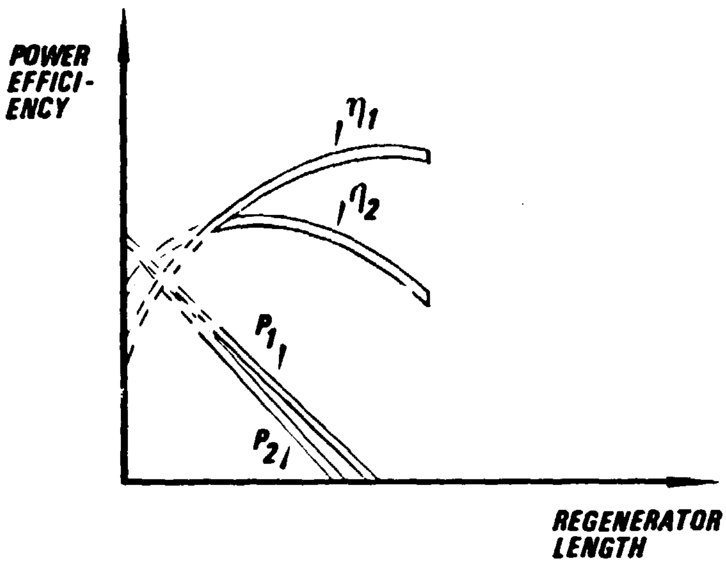
FIG 23: Principal diagram, comparison between different types of Stirling engines

2.4.3 Regenerator analysis

Two different types of regenerators have been analyzed by computer: separate and annular.

In general the regenerator matrix mass needs to be relatively large for two reasons: to get a high energy storage capacity; and to provide sufficient heat transfer area (small wire diameter of the matrix core) to transfer the stored heat to the gas. A large diameter cylinder/regenerator housing as required for an annular configuration will mean thick walls. Thick walls will, however, also cause more heat conduction losses.

Other parameters, such as filling factor and regenerator length, will also influence performance (shown in fig 24).



1,2 = DIFFERENT FILLING FACTORS

$f_1 < f_2$

FIG 24: Regenerator analysis, parametric study

Computer optimization of the engine will automatically dimension these parameters.

A theoretical analysis shows that one single or several smaller regenerators will have the same heat conduction losses. Design considerations will, however, give an optimum number of regenerators.

Not only heat conduction losses of the regenerators shall be taken into consideration when designing the engine, also the heat conduction losses from the cylinders shall be included. This will give the result shown in fig 25. When designing the various cylinder/regenerator arrangements of the different engines, the results have to be considered more in detail.

The performance of the regenerator depends on crossection area, length, wire diameter and matrix filling factor. Optimum performance from efficiency point of view occurs at different lengths depending on matrix filling factor. Power will always decrease if length increases. The optimization computer program takes all the parameters into consideration when calculating the performance at the design point.

2.4.4 Cylinder/regenerator arrangement

The cylinder (s)/regenerator (s) can be arranged in different ways.

The regeneratoıs can be either separate or annular and the number of separate regenerators can vary from one to several per cylinder.

The dimensions of the regenerator- height, diameter, filling factor, wire diameter - will influence the performance and it is very important to optimize the regenerator carefully. This is done as part of the general computerized engine optimization.

For example:

- two regenerators with different filling factors will have their optimum efficiency (not the same value) at different regenerator lengths.
- the output power will, however, decrease if the length increases.

For the single acting engines there will be one single cylinder. In a two cylinder engine the working cylinder will not influence specifically the cylinder/regenerator arrangement.

For the double acting engines the number of cylinders will be 4.

The single acting engines can have either an annular regenerator or six to twelve smaller regenerators surrounding the single cylinder in a symmetrical fashion.

The double acting engines can have either an annular regenerator per cylinder or one to three regenerators per cylinder.

In the filling factor* of the matrix of an optimized regenerator is increased, the flow losses will increase faster than will cycle improvements due to better heat transfer and heat storage, and therefore power will decrease. However, improved heat storage in the regenerator will increase cycle efficiency, but not necessarily brake thermal efficiency.

When designing the cylinder/regenerator arrangement using optimized dimensions limitations in pressure vessel design may set an upper limit for pressure vessel diameter. Fewer cylinders mean larger cylinders, and large cylinders results in thick walls. This will cause high bending stresses in the walls and make manufacturing very difficult.

The choice of annular regenerators versus separate regenerators is influenced by the above trade-off parameters.

Fig. 25 and 26 provide some general trends of losses as a function of the number of regenerators.

* $\dfrac{\text{vol. of metal in matrix}}{\text{vol. of empty matrix}}$

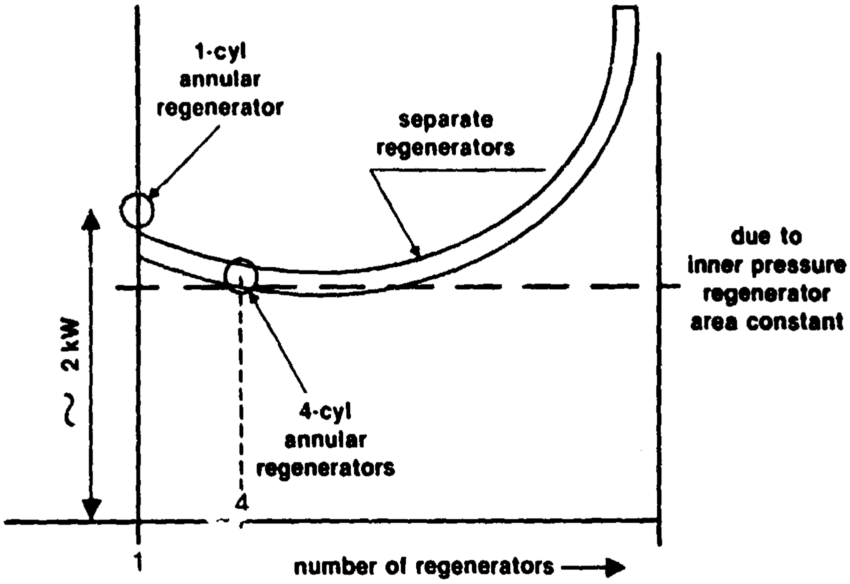


FIG 25: Heat conduction losses through cylinder and regenerator housings Comparison of concepts.

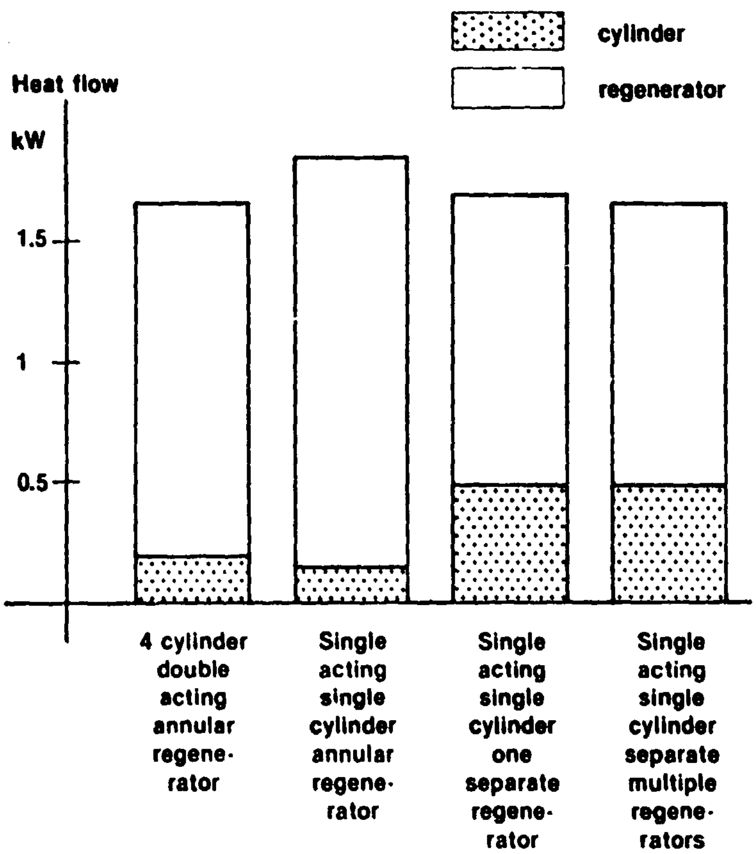


FIG 26: Heat conduction losses through cylinder and regenerator housings. Comparison of concepts.

2.4.5 Dead volume analysis

When designing an engine it is very important to minimize the total dead volume. Figure 27 illustrates the general effects.

Note that any increase in dead volume for a given engine displacement decreases power. A small increase in dead volume, however, can actually result in a slight gain in thermal efficiency, as can be seen in the figure. This effect applies only to the cold spaces of the engine. Any increase in dead volume in the hot spaces, either between the regenerator and the heater or between the cylinder and the heater, will always decrease the efficiency and power.

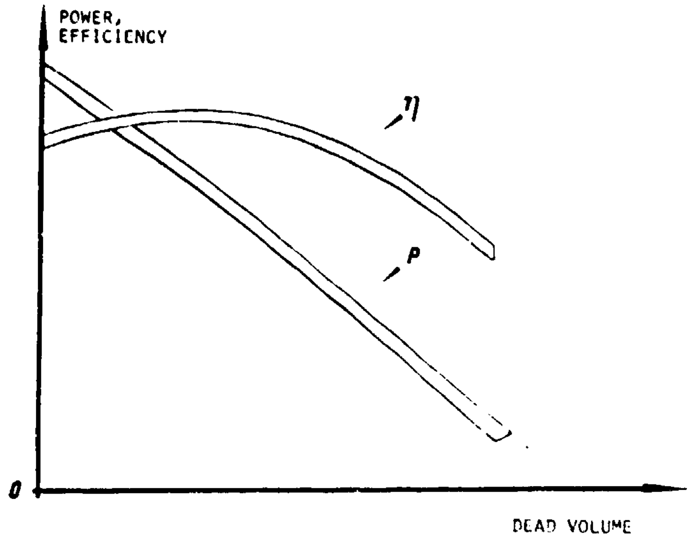


FIG 27: Dead volume analysis. Performance at various dead volumes.

2.4.6 Seal system

The seal system of the Stirling engine has two functions:

1) Keep the pressurized working gas in the cycle and seal it from leaking to the atmosphere.
2) Preventing oil from migrating into the Stirling cycle.

The seal system can be either of diaphragm or rollsock type in combination with a seal keeping the pressure difference between working gas and the atmosphere or of sliding type, which is a combination of both gas and oil seal. The diaphragm seal (rollsock) is an absolute barrier where no oil can pass through and into the cycle.

The diaphragm seal (rollsock) can be used in combination with the sliding seal to fully ensure oil from migrating into the cycle within the life requirements which are specified for the engine in a solar application.

All of the Stirling engine types, whether single-acting or double-acting, are able to use any of the seal systems. However, the single-acting engine has a somewhat limited space within the power piston for a seal (displacer rod), compared to the double-acting types. In addition, the seal in the piston is subject to high inertia forces any may be difficult to lubricate (where required) or to scavenge excess lubricant, in comparison to a stationary seal capsule. It is also more difficult to cool a reciprocating seal.

2.4.7 Drive system

For a single cylinder engine the rhombic drive offers the best solution because it provides complete balancing (including the higher harmonics) and a convenient means to achieve a nearly optimum phase relationship between the cold and hot volumes in a relatively compact mechanism. It also performs the function of a cross head as well as permitting a buffer space under the power piston which allows the crankcase to operate at atmospheric pressure.

Two cylinder single-acting engines can use either a rhombic drive, a V drive or a U drive. Double-acting engines can use a V, a U or a nutating drive (swash plate, Z crank, etc.).

Based on scattered calculated and measured data, friction losses for a single- and a double-acting engine for the same output power can be comparatively evaluated as follows:

FP = friction power
sa = single-acting
da = double-acting

Basis	Friction Power Relation
Miscellaneous Data	$FP_{sa} = 1.1 \times FP_{da}$
USS Emperical Formula	$FP_{sa} = 1.3 \times FP_{da}$
Assumed Difference	$FP_{sa} = 1.2 \times FP_{da}$

FIG 28: Friction losses

The evaluation is very rough but does indicate the difference between the two engine types. The double-acting engine has the lower friction losses.

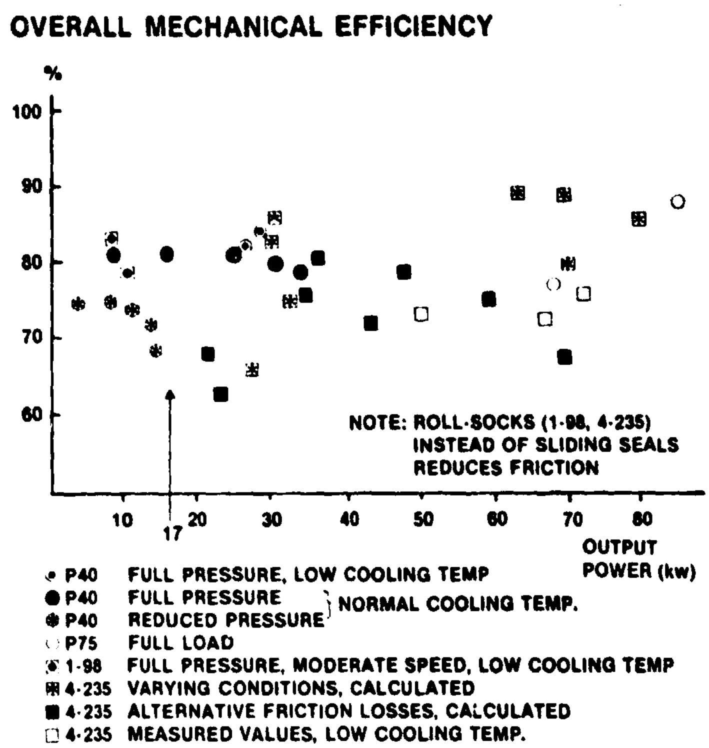


FIG 29: Mechanical efficiency for drive,piston rings and piston rod seal. Comparison of engines

Figure 29 presents the overall mechanical efficiency vs. power of four different Stirling engines, two rhombic drive machines (1 - 98, 4 - 235) and two double-acting types, the P40 and the P75. It is hard to draw absolute conclusions, particularly since mechanical efficiency is difficult to measure accurately on all piston type engines. It is apparent, however, that the single-acting 1 - 98 and 4 - 235 engines have the advantage of lower seal friction losses since they were designed for operation with rolling-sock seals. Placing rolling-sock seals on the P40 ans P75 would have raised their mechanical efficiency approximately 3 to 4 percentage points. The rolling-sock seal is only of academic interest, however, since it has proven to be unreliable and would not be seriously considered for either type of engine.

2.4.8 Control principles

The Stirling engine can be controlled in several ways to get varying output power. The engine can be controlled by means of

- mean pressure control
- pressure amplitude (dead volume) control
- engine stroke control
- phase control (phase between pistons)
- temperature control (working gas temperature)

For the solar application either mean pressure or pressure amplitude control are most suitable. Stroke control is more or less limited to nutating drive mechanisms and involves a complex and relatively heavy control system. Phase control also involves complex and heavier drive systems which are more bulky and difficult to balance.

Based on considerations of efficiency as shown in fig 30 mean pressure control was recommended. The pressure amplitude (or dead volume) control system consists of a number of varying sizes of vessels which makes the design more complex than the mean pressure control system, including one buffer and a compressor.

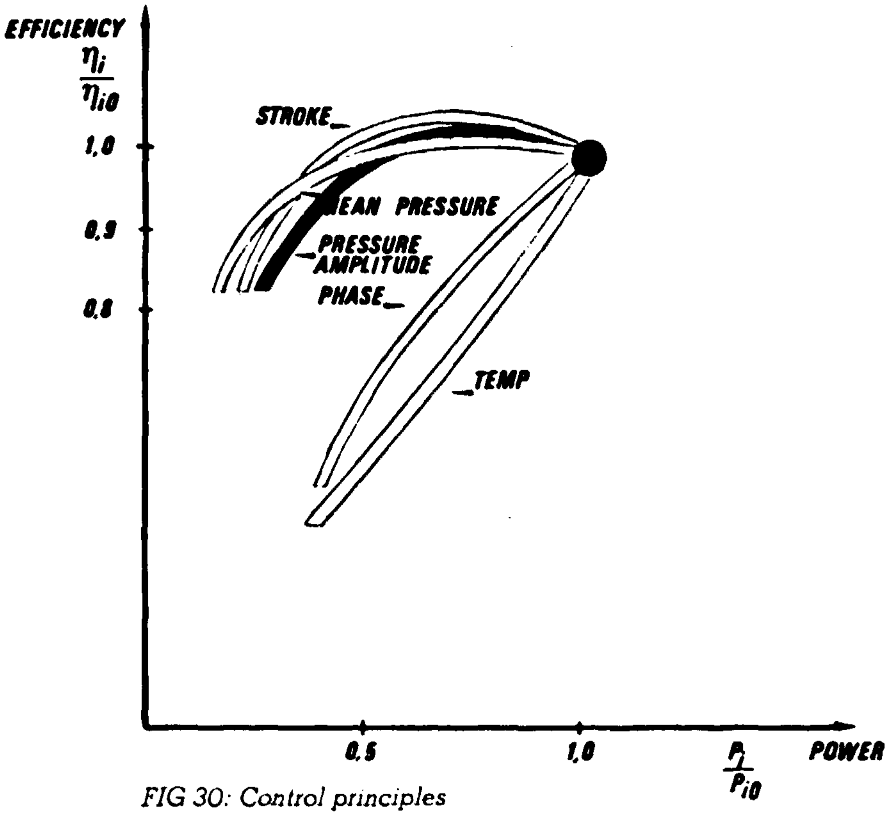


FIG 30: Control principles

2.5 Concept evaluation

2.5.1 Selection of concepts

After a more thorough evaluation and after having made lay-outs of a number of concepts the following engine configurations were selected for more detailed design and analysis. The basis for the selection includes cycle performance and efficiency, amount of dead volume and buffer needs, heater design details, regenerator configuration, seals and rings, balancing and flywheel requirements, drive mechanism forces and number of parts, friction, manufacturing cost, scaling rules and estimates of reliability.

1) four cylinder double acting engine with multiple (two per cylinder) regenerators and U-drive

2) four cylinder double acting engine with annular regenerators and U-drive

3) single acting one cylinder engine with multiple regenerators and rhombic drive

4) single acting two cylinder engine with multiple regenerators and conventional crank shaft.

2.5.2 Comparison of concepts

The table (fig 31) shows a comparison of concepts, components and other factors taken into consideration.

The different factors have been compared one by one, not bringing the whole engine into the evaluation.

Object/Component	4-cyl, Double acting	1-cyl, Single acting Rhombic drive	2-cyl, Single acting U or V-drive
Cycle performance, efficiency	The cycle efficiency is approximately equivalent. However, a tendency towards lower efficiency exists for the single acting type due to slightly higher parasitic losses. (see section 2.4.7)		
Dead volume	The double acting engine will have a slightly bigger dead volume on the cold side than the single acting engine. The dead volume on cold side will, however, not decrease performance.		The dead volume needed to connect the cylinders do necessarily yield an engine with lower specific power. In addition there will be some extra flow losses in the connecting ducts.
Buffer volume	Not needed	Needed	Needed
		The buffer volume is needed to balance the forces to the drive mechanism and may be chosen either at min pressure or mean pressure. No buffer volume lead to excessive forces. Buffer volumes yield additional pumping- and heat transfer-losses.	
Heater	Heater tube distribution to cylinders and regenerators not always symmetrical.	The heater design needs additional development work for the single acting configuration in order to achieve comparable heat transfer numbers.	

Object/Component	4-cyl, Double acting	1-cyl, Single acting Rhombic drive	2-cyl, Single acting U or V-drive
Bellows	Must be applied but can be avoided if separate heat pipes will be used.	Most probably to be used to allow flexible joints at the cold connection ducts. Any design involving a rigid deck in this area tends to be rather complex. It is to be observed that a number of regenerator units must be used for single acting engines.	
Regenerators	4 or 8 separate or 4 annular.	Flexible but preferably 6 units per cylinder to achieve an even mass flow distribution of the working medium. (Annular not to be recommended because of stress consideration in a cylinder twice the size of the double-acting design)	
Seals	4	2	2
Pistons/piston rods	4	2	2
Balancing	Completely balanced.	Completely balanced.	Balancing shaft, needed to achieve satisfactory balancing.
Torque	Small torque variation. Positive torque at all crankangles.	Large torque amplitudes. Larger flywheel needed.	
Inertia and pressure forces on drive system		Relatively large	Relatively large
Drive mechanism	2 crank shafts 4 connecting rods 1 power take-off shaft 4 cross heads	2 crank shafts 6 connecting rods 2 yokes	1 crank shaft 2 balancing shafts 2 connecting rods 2 cross heads
Mechanical friction		Slightly larger than for comparable 4-cyl double acting, due to larger bearings etc.*	Larger than for the 4-cyl double acting version.
Manufacturing cost	Preliminary cost evaluation with the aim of comparison between these alternatives for automotive applications, show a cost advantage for the 4 cylinder double acting version.		
Scaling	Scaling rules applicable to 600 kW	Scaling up to 500 kW output of a 1-cyl single acting engine may provide some uncertain factors. Regenerator location tends to be a limiting factor.	
Reliability	More dynamic components may be a disadvantage.	The complexity of the rhombic drive, the larger forces acting etc must be investigated further.	Excessive forces and relatively large dimensions of components may provide some uncertainties with respect to reliability.

FIG 31: Comparison of concepts

The conclusion of the table is that there are only small differences between the engines. The significant differences are:

- buffer volume
- seals
- torque
- heater/bellows
- drive mechanism
- scaling

*Piston ring friction tends to be nearly equal for geometrically similar rings if the total piston area for different engines designs is equal. For example a single piston engine having the same area as a 4 cylinder engine, operating at the same mean pressure and the same piston speed, should have nearly equal piston ring friction since the ring face area for the two engines should be equal.

2.6 Lay-out of concepts

2.6.1 Optimization and analysis

The different concepts were first optimized without any design restrictions. The United Stirling optimization computer program operates with 20 optimization parameters. Upper and lower limits can be set for these parameters. The optimization criterion is maximum efficiency at required power level.

However, the Stirling engine design has a number of geometrical restrictions — for example the connections from cold to hot side are both across piston and piston dome and across heater, regenerator and cooler — which should be included in the optimization. Of course a computer code can be made, which will take this into consideration when optimizing the engine. Such a code will, however, be very complex and we have found that it is better to optimize with some fundamental restrictions, make a rough lay-out based on the optimization results and then make a new optimization with further restrictions. This iterative process is continued until a satisfactory design is reached.

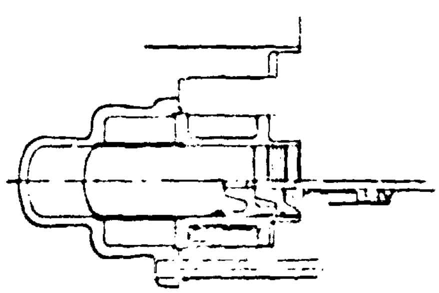
The resulting engine designs are shown in separate drawings. Data and comments on optimized designs are listed below for the selected concepts.

2.6.2 Four cylinder double acting engine, annular regenerators. Indirect heating; U-drive (fig 32, 33)

Basic data

Bore	48 mm	
Stroke	40 mm	
Swept volume	73 cm^3	
Regenerator	Cross section area	3970 mm^2/cyl
	Diameter	89 mm
	Height	50 mm
Heater	Number of tubes	15/cylinder
	Tube dimensions	3 mm inner diameter
		4.5 mm outer diameter
		200 mm length
Cooler	Number of tubes	460/cylinder
	Tube dimensions	1 mm inner diameter
		2 mm outer diameter
		75 mm length

The engine is a four cylinder double-acting design as shown in figure 32 and 33, with twin crankshafts geared to a central output as in the P40 engine. Regenerators are incorporated in an annular ring surrounding each cylinder rather than in separate housings as typified by the P40 design. A heat pipe heater head is illustrated, with thin flexible sheet metal (bellows) around the cylinder heads to permit thermal expansion and seal the sodium chamber. The heater tubes are arranged symetrically about the cylinder center line, which is ideal for proper gas distribution to the regenerator.



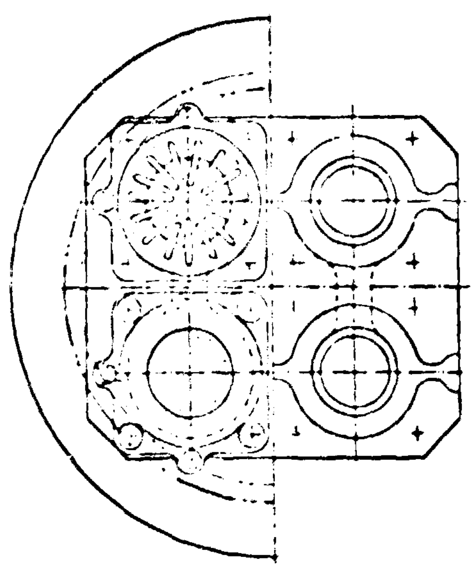


FIG 32: Four-cylinder double-acting engine, annular regenerators — top view

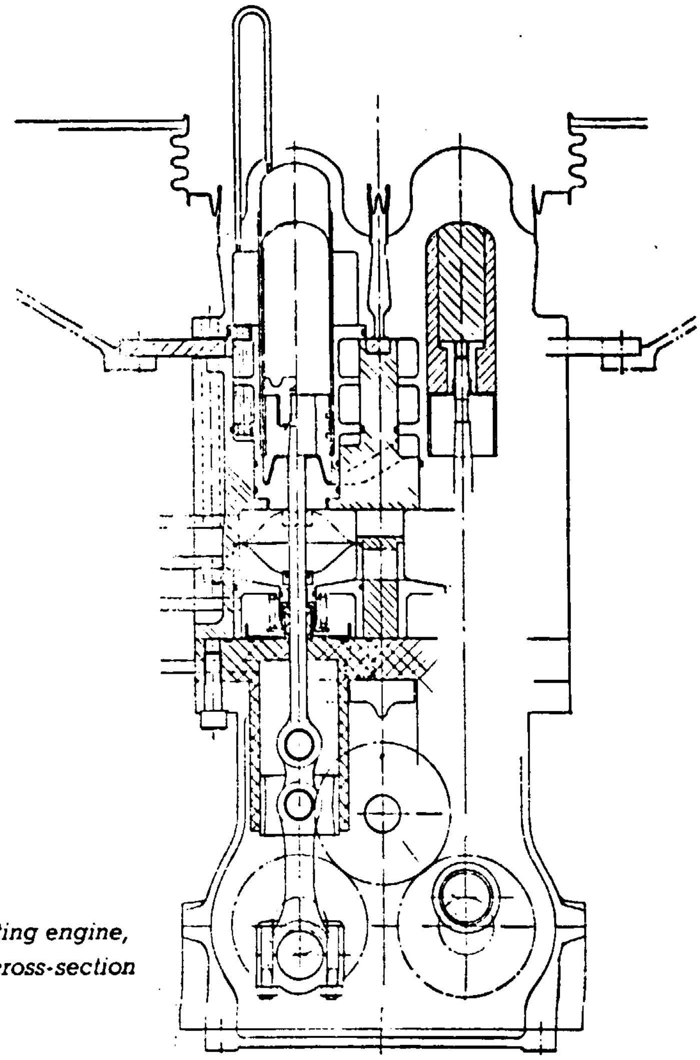


FIG 33: Four-cylinder double-acting engine, annular regenerators — cross-section

2.6.3 1 cylinder single acting engine, multiple regenerators. Indirect heating, rhombic drive (fig 34)

Basic data

Bore	96 mm	
Stroke	Radius 20 mm, rod 66	mm
Swept volume	318 cm^3	
Regenerator	Cross-section area	14900 mm^2
	Number	12
	Diameter	40 mm
	Height	45 mm
Heater	Number of tubes	48
	Tube dimensions	3.7 mm inner diameter
		5.5 mm outer diameter
		247 mm length
Cooler	Number of tubes	1714
	Tube dimensions	1.0 mm inner diameter
		2.0 mm outer diameter
		85 mm length

The engine is a one cylinder, displacer engine with rhombic drive and separate regenerators similar to the Philips 1-98 engine design.

The design includes 12 regenerators to get the simplest interface between cylinder/regenerator arrangement and a heat pipe. The design — as shown in figure 34 — includes no buffer volume, but the engine requires one for operation.

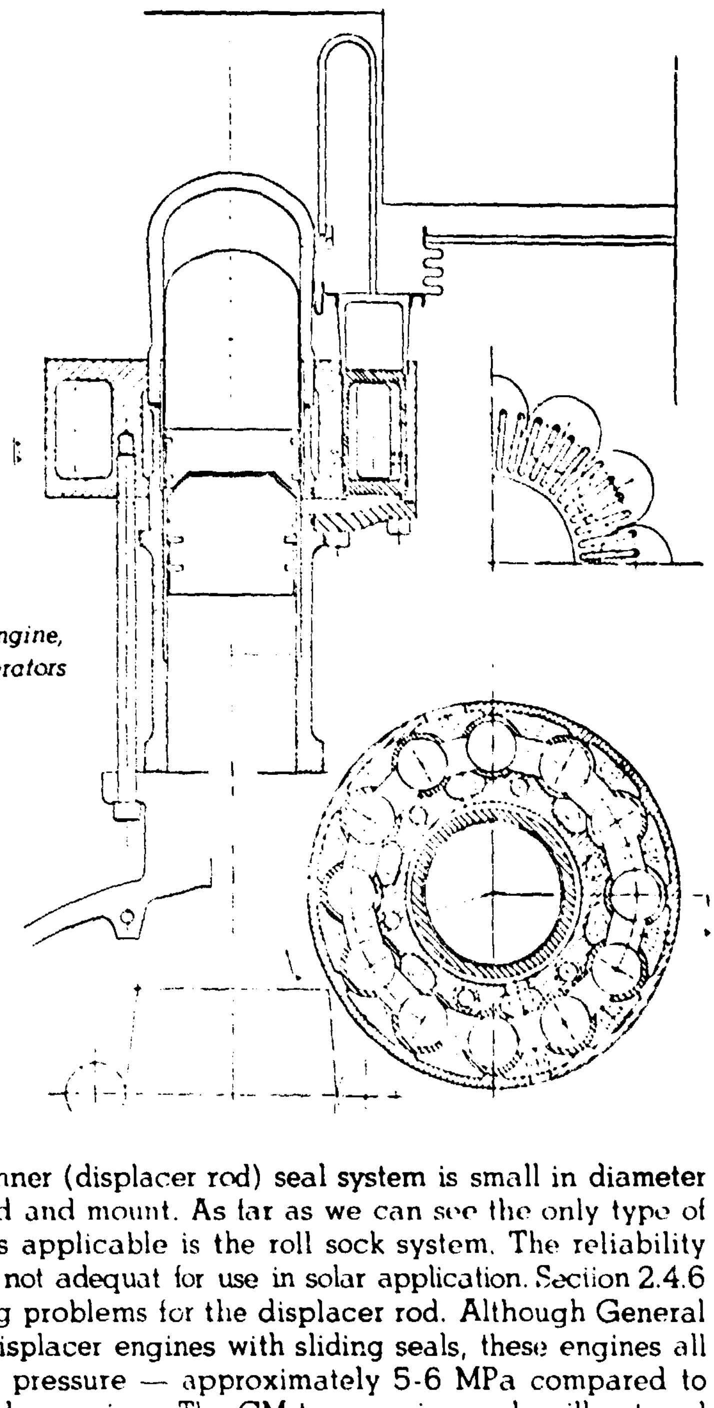


FIG 34: Single cylinder single-acting engine, multiple regenerators

The rhombic drive inner (displacer rod) seal system is small in diameter and difficult to build and mount. As far as we can see the only type of seal system which is applicable is the roll sock system. The reliability of this seal system is not adequat for use in solar application. Section 2.4.6 discusses the sealing problems for the displacer rod. Although General Motors built many displacer engines with sliding seals, these engines all operated at modest pressure — approximately 5-6 MPa compared to 15 MPa for present day engines. The GM type engine seals will not seal adequately at this pressure level, and their wear rate is too rapid.

Figure 35 shows an optional design of a housing for a single cylinder with an annular regenerator. To get enough regenerator area the outer diameter of the housing has to be increased considerably compared to the basic design. This makes the walls very thick, which makes the housing difficult to manufacture.

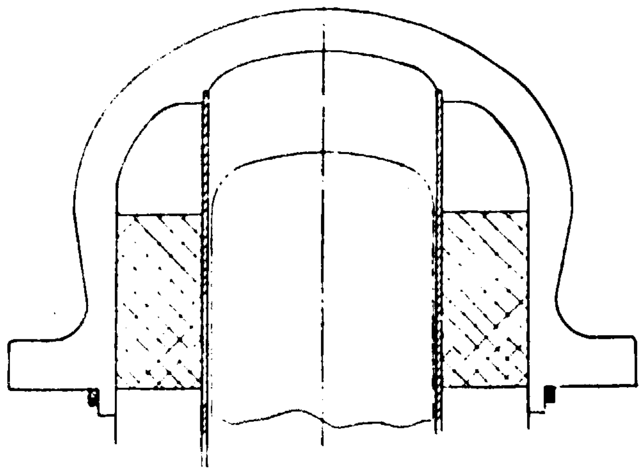


FIG 35: Alternative cylinder/regenerator housing design, single cylinder engine — annular regenerator

2.6.4 2 cylinder single acting engine, multiple regenerator indirect heating. Single crank shaft (fig 36)

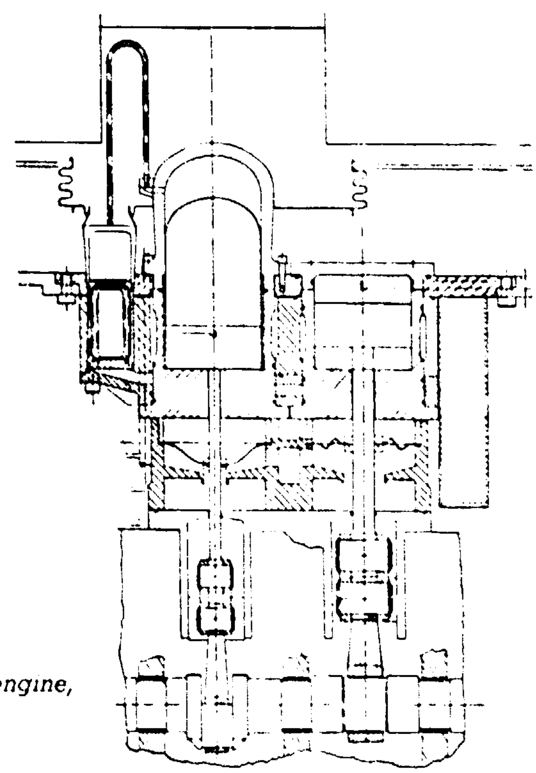


FIG 36: Two cylinder, single-acting engine, multiple regenerators

Basic data

Bore	96 mm (displacer cylinder)	
	96 mm (working cylinder)	
Stroke	44 mm	
Swept volume	318 cm^3	
Regenerator	Cross section area	12500 mm^2
	Number	10
	Diameter	40 mm
	Height	45 mm
Heater	Number of tubes	40

The difference between this engine and the single cylinder engine is the separation of the displacer cylinder and the working cylinder. The cold spaces are then divided between two cylinders, resulting in larger cold volume dead space than for displacer type designs.

The drive system no longer has to be rhombic. This configuration can use the conventional type of crank shaft with cylinders placed in a V or in line.

The number of regenerators have been decreased compared to the previous design to permit location of the two cylinders close to each other for minimum dead volume in the cold connecting duct.

Seal systems of any suitable type can be used in this two cylinder engine. If the diaphragm seal system is used a third dummy cylinder must be included in the design to keep gas volumes on either side of the diaphragm constant. The buffer volume is shown in the design.

2.6.5 Engine heater head designs for alternative solutions of indirect and direct heating

A number of heat pipe designs have been performed. Sketches and drawings showing the various heater head designs and adaptation to the heat pipe are shown in the following figures.

Fig. 37 offers a very simple design of the heat pipe. The heat pipe has no thermal storage. Joining to the engine heater head is by bellows and a corrugated plate between the cylinder and regenerator housings. Wicking material would be placed on the inside of the solar receiver surface as well as along the side walls.

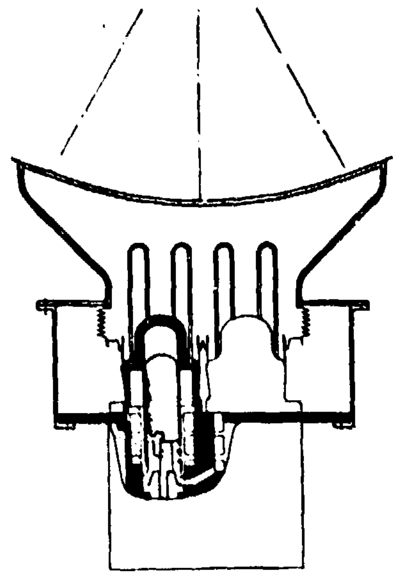


FIG 37: Doble-acting advanced solar engine, annular regenerators and a simple heat-pipe with no thermal storage.

Fig. 38 shows the Stirling engine mounted on a heat pipe heat receiver with a thermal storage. The design was made by General Electric under contract to JPL.

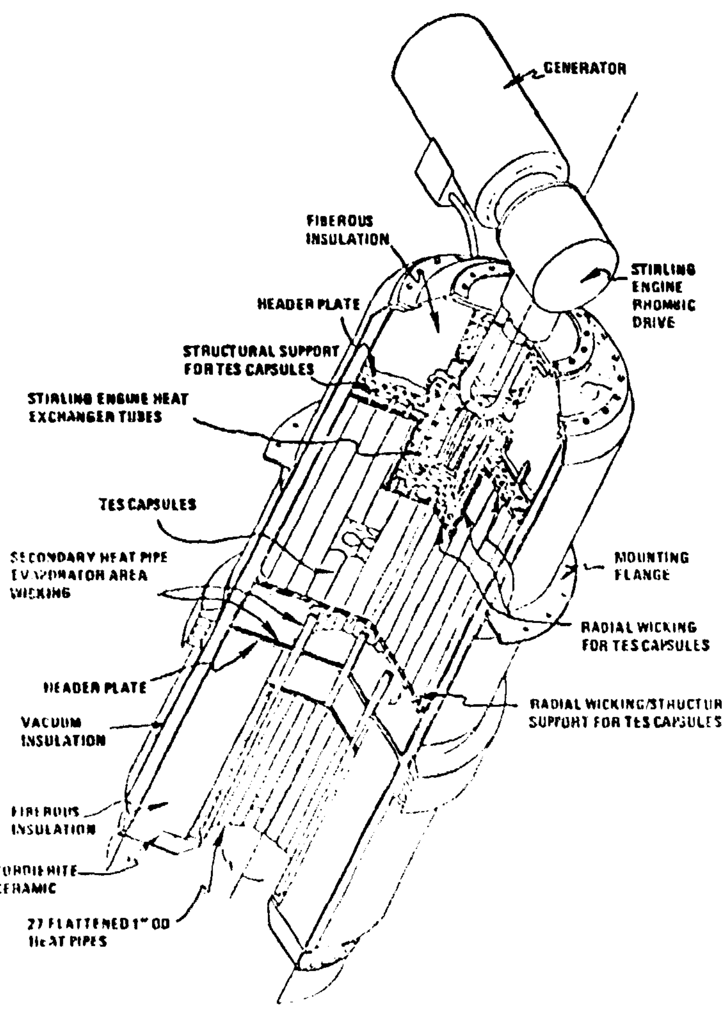


FIG 38: Focus-mounted heat-receiver TES/Stirling engine-regenerator (from 1st semi-annual solar meeting report)

Fig. 39 shows an alternative to the GE receiver also including thermal storage. The design shown has only one primary heat pipe including the thermal storage (compare the GE receiver having one primary heat pipe and one secondary heat pipe including the thermal storage).

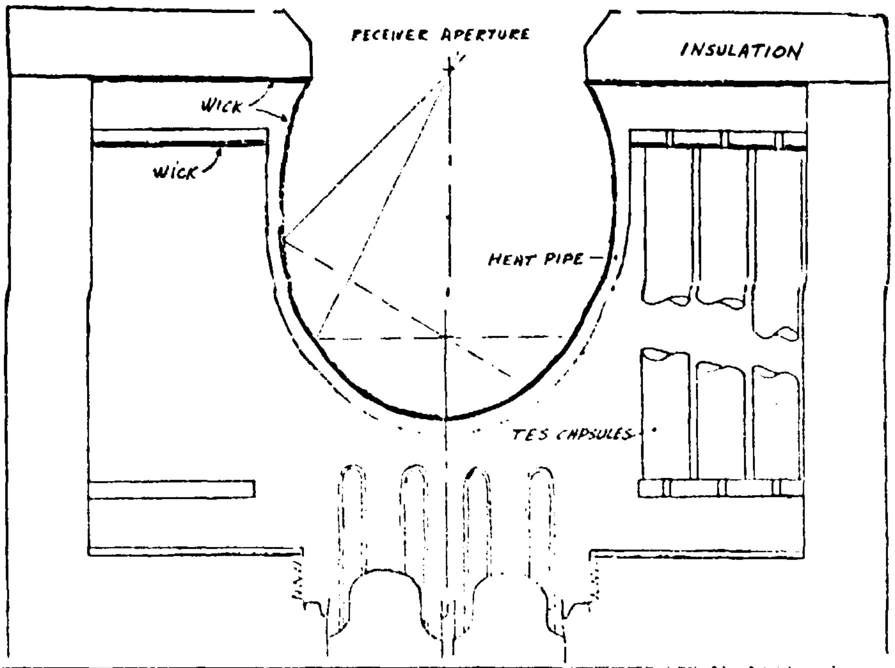


FIG 39: United Stirling design of a heat-pipe and thermal storage

Appendix 1, IECEC paper 739073, summarizes previous United Stirling experience of joining heat pipes with Stirling engines.

As described earlier in this report, the heater tube design of a heat pipe heater can have any shape. The design shown here incorporates a number of small straight tubes connecting cylinder and regenerator housing.

Fig. 40 and 41 illustrates some very early design ideas for direct insolation. They are only rough lay-outs and have not been evaluated in detail. The design principle is to reradiate insolation from a ceramic body. The heater tubes will thus receive heat on both sides of the tube — on the front side directly from the insolation, on the back side via reradiation from the ceramic body. This will improve heat transfer but may cause increased receiver losses.

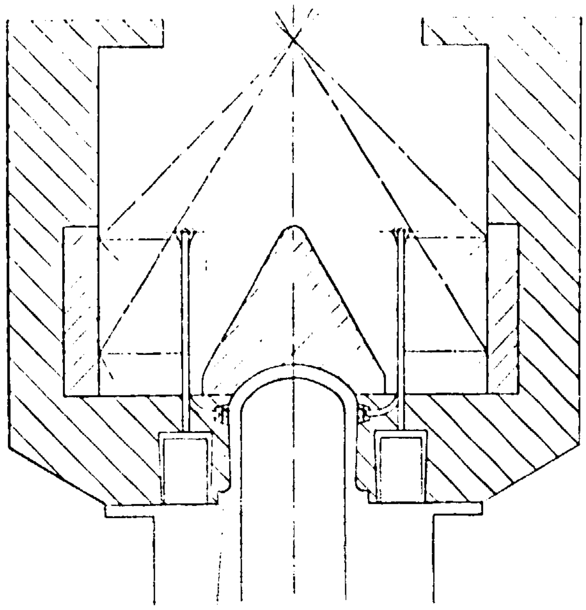


FIG 40: Heater design for direct insolation using secondary radiation

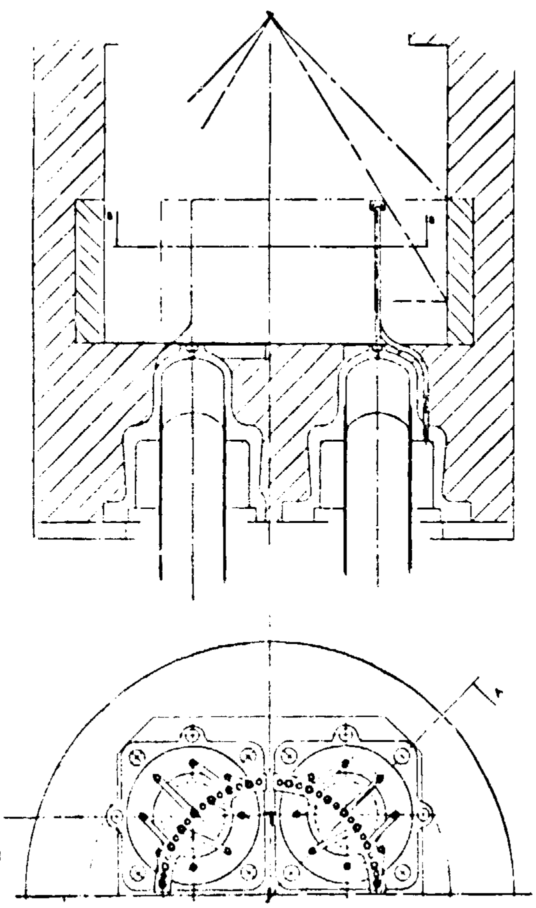


FIG 41: Heater design for direct insolation using secondary radiation

2.7 Performance

2.7.1 Basis

The method of evaluating the performance of the advanced concepts consisted of utilizing the USS computer program, which has been calibrated and validated by several engine generations of United Stirling and more recently be extensive P40 engine experience. In addition special tests have been performed on P40 engines in United Stirling's laboratory to simulate solar power operating conditions.

Tests with helium and hydrogen as working gases in a P40 engine were performed to study performance at different load and engine speeds. The results are in good agreement with computed values.

Tests on heaters at different temperatures were also performed to evaluate the influence of temperature variations and to study the heat losses at higher temperatures. Calculations and measurements correspond well.

Tests were also made with different starting methods

- starting with limited heat supply
- starting (cranking) at different temperatures on heater

2.7.2 USS P40 engine

The P40 engine is a Stirling engine designed in 1975 at United Stirling. Tests with the P40 engine have been performed in the laboratories since 1976.

The P40 engine is a 4 cylinder engine, of U-type with two crank shafts geared to a central output shaft. The engine has 8 regenerators (2 per cylinder) and the heater is of involute type (the tubes connecting cylinder and regenerators have an involute shape). Figure 42 shows the design.

The proposed double acting advanced solar engine design is rather close to the P40 engine; the drive system of U-type, 4 cylinders, seal system of sliding type, control system of mean pressure type.

The regenerators are different — the advanced engine is of annular type, and the heater design for the solar engine will meet the assumptions of the insolation power.

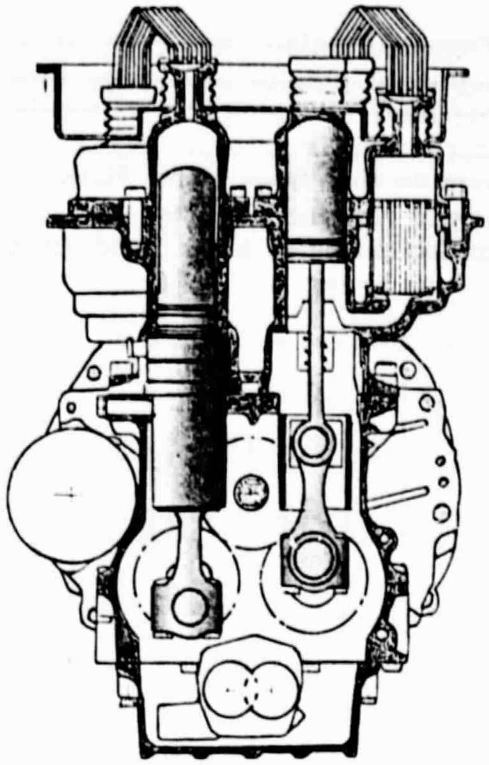


FIG 42: Principal design of four cylinder double-acting engine, two separate regenerators per cylinder.

A 20 kW advanced solar engine will have almost the same main dimensions as the P40 engine and the output power of the two engines will be very close to each other at the same rpm.

2.7.3 Prediction of performance of advanced concepts

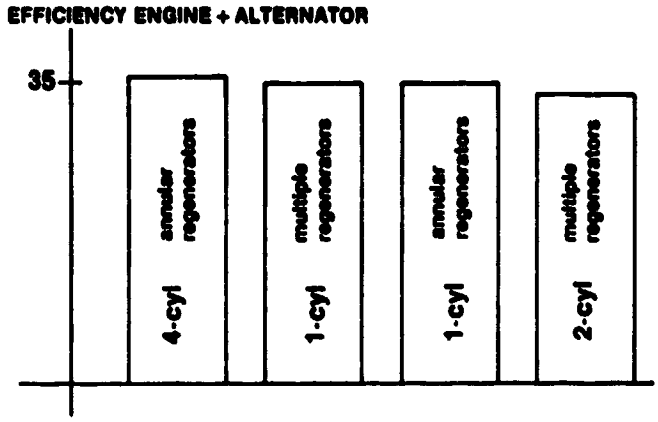
The different engine performances were calculated with the United Stirling computer programs based on the present data for the P40 engine.

These programs are valid for existing engines and include emphirical correction factors for adjusting computed values to measured. The calculation of performance of the engines will therefore give today's performance and not performance of the advanced engines. The improvements of the advanced engine are to be accounted for separately.

No significant difference in cycle performance of the different engine designs exists.

The mechanical efficiency is higher for the double acting engine than the single acting engines which are projected to have higher losses in the drive system. The single acting engine must be designed to meet the higher piston forces due to the working principle. The efficiency of the alternator and auxiliaries are the same for the different engines.

From the performance comparisons of the figures 43 and 44 (which represent present technology level) it can be seen that, within the accuracy of prediction currently available, the efficiency of all four designs is so nearly equal that it is difficult to make a final selection on that basis alone. However, the two cylinder version has the lowest efficiency, and because of its other disadvantages, such as balancing problems, larger dead spaces and greater engine bulk, it would not be recommended for further consideration (see 2.8).



	INDICATED ENGINE EFFICIENCY	MECHANICAL EFFICIENCY	TOTAL EFFICIENCY
4-cyl ann. regen.	51.0	0.80	40.8
1-cyl mult. regen.	52.0	0.76	39.5
1-cyl ann. regen.	52.0	0.76	39.5
2-cyl mult. regen.	50.0	0.76	38.0

Ambient Conditions:

cooling water temp.	43°C
heater head temp	800°C
working gas	Helium
electric output	15 kWe

FIG 43: Performance comparison of different engine types. Engine indicated efficiency is defined as: $100 \times \frac{\text{gross output (no mechanical friction)}}{\text{net heat input to heater}}$

Total efficiency, which is the product of indicated efficiency and mechanical efficiency, does not include accessory losses such as pumps, fans etc.

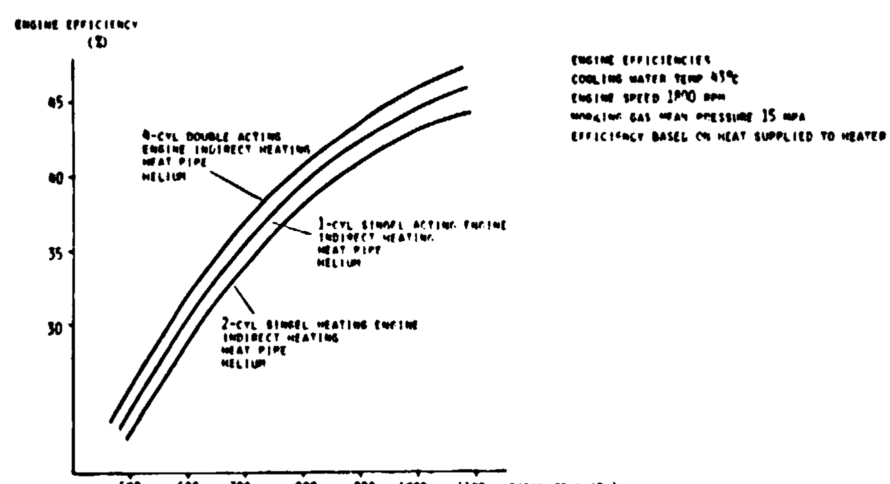


FIG 44: Performance comparison of different engines at various heater temperatures

By 1985, however, a stepwise improvement of double-acting engine performance is expected to be realized. This is based on component improvements as well as a better understanding of solar optimization and its implementation into the engine hardware, principally the solar receiver, regenerators, coolers and passages.

The following component improvements are projected;

- drive system
  smaller dimensions, improved power transmission
- seal system
  decreased seal power losses, minimized pumping losses
- heater
  minimized flow losses, improved temperature distribution, minimized dead volumes
- regenerator
  improved design, improved flow distribution
- cooler
  improved cooling water flow distribution
- cylinder and regenerator housings
  improved design, decreased heat conduction losses
- cycle
  improved cycle design, minimized losses

The improvements are expected to increase the engine efficiency above the baseline by about 3 efficiency points resulting from minimizing the parasitic losses.

The efficiency of the advanced engine is expected to be above 45% and if new materials (as for example ceramics) can be imcorporated, allowing higher temperatures the engine efficiency could be above 50% (fig 45)

A discussion of use of ceramics in the advanced solar engine concept for further performance improvements will be found in the appendix.

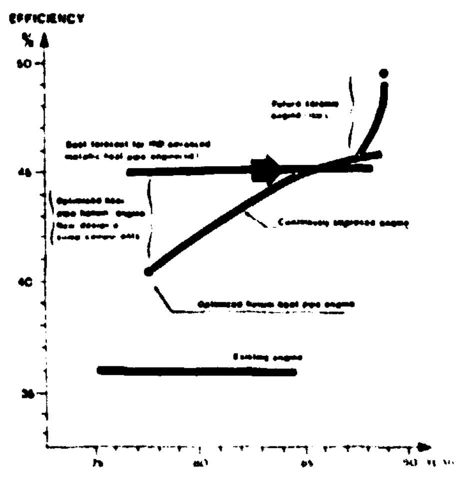


FIG 45: Prediction of performance, advanced, kinematic solar Stirling engine

2.8 Engine selection for conceptual design task

After completing the configuration definition and the analysis of the configurations studied, a ranking procedure was employed as a concept means of identifying the most viable and promising. The ranking is shown in fig 46. Note that concepts which are given the same rank are enclosed by brackets.

Code:
4 cyl, multiple reg I
4 cyl, annular reg II
1 cyl, single acting III
2 cyl, single acting IV

Ranking factors	Ranking order	Notes
Cycle performance, efficiency	III, II, I, IV	Almost same cycle efficiency. Cf flow losses, mechanical losses, heat conduction losses, dead volume losses.
Dead volume	II, I, III, IV	Two cylinders have a big cold connecting duct.
Buffer volume	(I, II), IV, III	Double acting needs no buffer.
Heater	II, III, IV, I	Many heater tubes per cylinder complex.
Bellows	II, III, IV, I	Separate regenerators for 4 cyl engines needs bellows. Other, some type of metal plate.
Regenerators	II, IV, III, I	1 regenerator per cylinder has an upper cylinder diameter limit. Annular type can be an advantage.
Seals	IV, I, II, III	1 cyl single acting seal system not solved.
Balancing	III, I, II, IV	Rhombic drive completely balanced. 2 cyl engine need balance shaft.
Torque	I, II, III, IV	Big torque variation for single acting engine. Flywheel needed.
Forces	I, II, IV, III	Design must be made for max forces. Higher forces for single acting.
Drive mechanism	IV, I, II, III	2 cyl concept, normal crank shafts most simple. Rhombic drive complex.
Mechanical friction	I, II, III, IV	Slightly bigger for single acting due to design for max forces — parasitic losses higher.
High system efficiency	I, II, III, IV	Slightly lower for single acting due to mechanical efficiency and buffer volume losses.

Ranking factors	Ranking order	Notes
High system availability	IV, II, I, III	Number of components — hot parts, seal, drivesystem, etc.
Controlability	I, II, III, IV	Torque variations for single acting engine.
Simplicity	IV, II, I, III	Number of components, new designs, etc.
Low risk	I, III, IV, II	No experience of annular regenerators. No present experience of single acting engines and rhombic drive.
Maintenance	IV, II, III, I	Few components an advantage.
Life time	I, II, III, IV	No significant difference.
Well proven materials	I, II, III, IV	No significant difference.
High performance potential	II, I, III, IV	Few cylinders have an upper limit. Annular regenerators volume and flow distribution an advantage. Two cylinder concept has extra losses.
High insolation availability	I, II, III, IV	No significant difference.
High power potential	I, II, IV, III	Few cylinders have an upper limit.
Low cost components	IV, I, II, III	Annular regenerators and rhombic drive more complex.
Weight and installation	II, I, III, IV	Single acting engines bigger and have higher weight.
Volume	II, I, III, IV	Single acting engines have larger dimensions.

Fig 46: Ranking of engines

2.9 Recommendation on the choice of solar engine concept

The conclusions arrived at after completing the study were that the calculated performance of the different engine types are equal within the accuracy of the calculations.

Also from design point of view the concepts are equal. Neither was it possible at the present to identify any significant differences in the technological risk involved in the different concepts. An exception was made for the single cylinder single acting engine for which the only feasible piston rod seal system which we can presently identify is the roll sock system. Based on our previous experience with roll sock seal systems we rated these as a high risk.

The choice of engine concept therefore should be founded on other factors. It is the opinion of United Stirling that for small power levels the single acting concept is suitable mainly because it could minimize the number of components. For larger power levels this becomes less pronounced which makes the double acting multiple cylinder concept more suitable. It seems that a power level of 15 kW may be approximately the level where a transition from one type of engine to the other is motivated. The final power level in Solar Engine applications based on a concentrator of the dish type should thus preferably be identified and allowed to influence the decision of what engine concept should be further developed.

The fact that during the latest 5 or 7 years development activities concerning single acting engines have been small, whereas a larger effort has been made to develop double acting engines, has given the double acting engine a lead over the single acting engine. Should the single acting engine concept be chosen for solar powered engines, considerable resources are required in order to close this gap before the engine technology can be pushed further than what characterizes the double acting engines of today.

Based on the above considerations and the ranking procedure the recommendation of United Stirling on the choice of engine concept for further study was the double acting 4 cylinder concept. This recommendation was accepted by the NASA project manager.

3 DETAILED DESIGN

Based on the results of the configuration definition studies and subsequent analysis, the following baseline for the conceptual design and its implementation assessment was selected.

Alternator output	20 kW
Alternator efficiency	95% (given by NASA)
Number of cylinders	4
Number of regenerators; type	4 annular type
Heater input temperature	1500°F
Coolant temperature	110°F
Efficiency target	40-45%
Heater head temperature	
Mal distribution	
Full power	± 50°F
25% power	± 100°F
Insolation profile	Lancaster, California

Seals
a) Seal package which includes the USS diaphragm seal
b) Seal package which includes the USS sliding seal

Heater head
a) Direct solar insolation heater integrated with receiver component as required for engine assembly
b) Heat-pipe-fed sodium vapor condensate heater

Operational mode — Constant speed — 60 Hz with load delivered to grid

Technology readiness of advanced components 1985

3.1.1 Definition of design work

The detail design work includes a more thorough design and analysis of the components for one of the concepts in the concept study — the four cylinder double acting engine with annular regenerators.

The detail design work mainly concerned the heater head, the cylinder/regenerator assembly, and the seal system.

A detailed analysis of the control principle of the engine also was performed.

3.1.2 Reference work

Reference work concerning a solar powered Stirling engine, the solar insolation heat flux distribution, collector and receiver performance has been going on parallel to this study, and parts of this work have been included in the design and have been a guide to us, when designing the advanced solar powered Stirling engine.

Calculations of the heat flux distribution for different collectors and receiver cavities were performed by JPL. Results show the distribution in different points inside the cavity where the heat exchanger shall be located. The heat exchanger was designed to meet the requirements of a uniform temperature distribution.

The flux distributions are presented in appendix.

Information concerning insolation profiles, heat pipes, ranking procedures, assessment etc has been received from MTI.

Different types of heater head have been designed by Fairchild — with milled channels in plates and with tubes imbedded in copper. The designs show new technology within the areas of new materials, welding and brazing techniques. The different types of heaters presented seem to be very promising for use in combination with a solar powered Stirling engine.

3.2 Engine optimization

The engine was optimized with the new requirements of power output from the alternator, 20 kW, instead of 15 kW as required earlier. In addition, indirect heating was selected for the optimization case.

The main output data area:

Bore	54 mm
Stroke	43 mm
Swept volume	99 cm^3
Heater tubes	12/cylinder
Tube dimensions	4/6 mm (id/od)
Tube length	280 mm
Heat flux	24 W/cm^2
Cooler tubes	608/cylinder
Tube dimensions	1/2 mm (id/od)
Tube length	60 mm
Regenerator area	3970 mm^2/cylinder
Regenerator length	70 mm
Wire dimensions	50 microns wire diameter
Filling factor	34%

The operating conditions stated for the optimization were:

Heater temperature	800 °C
Coolant temperature	43 °C
Working gas mean pressure	15 MPa
Working gas	Helium
Engine speed	1800 rpm
Indicated output power	27.5 kW

This data set has been used in the design of the engine for indirect heating. The data set has also been used for designing the "engine body", which is the same one for both indirectly heated engine and directly heated engine.

However, the data set for maximum efficiency cannot be exactly realized in the design due to geometrical and other design limitations, as for example, heater tube must fit between cylinder and regenerator (in spite of a demand for shorter tubes to maximize efficiency), height of cooler and regenerator must match cylinder height, and the cold connecting duct must connect cooler and cylinder.

3.3 Engine design

The detailed design of the engine was based on the optimized data.

The basis for the design was the 4 cylinder U-engine. The design work is the iterative process to match design requirements and optimization parameters — to redesign components to better match optimized data.

The design of one component is not an isolated problem, all other components are included more or less in the design.

The following will present the detail design work and the analysis of the components included in the design.

Drawings (fig 47, 48) show the design of the components and description of the components follows.

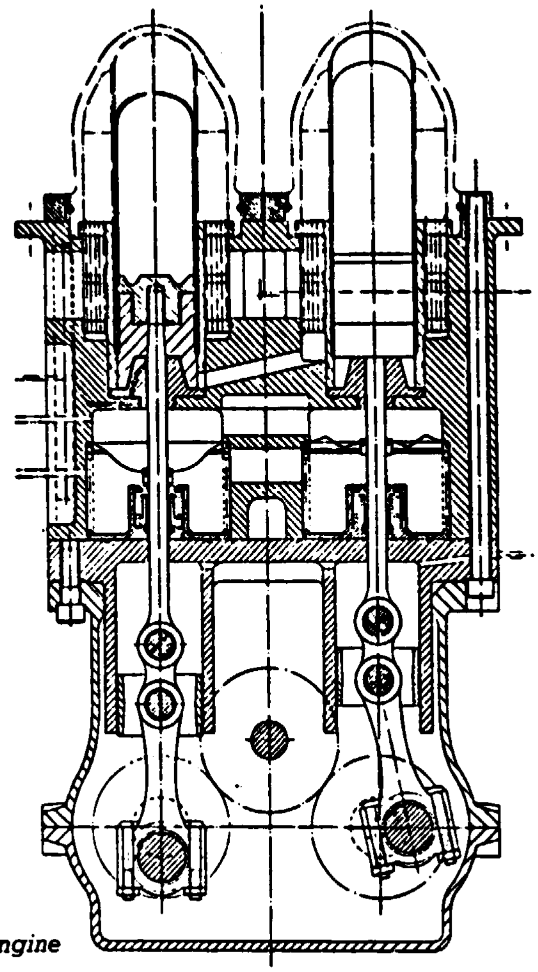


FIG 47: Cross-section, advanced kinematic Stirling engine

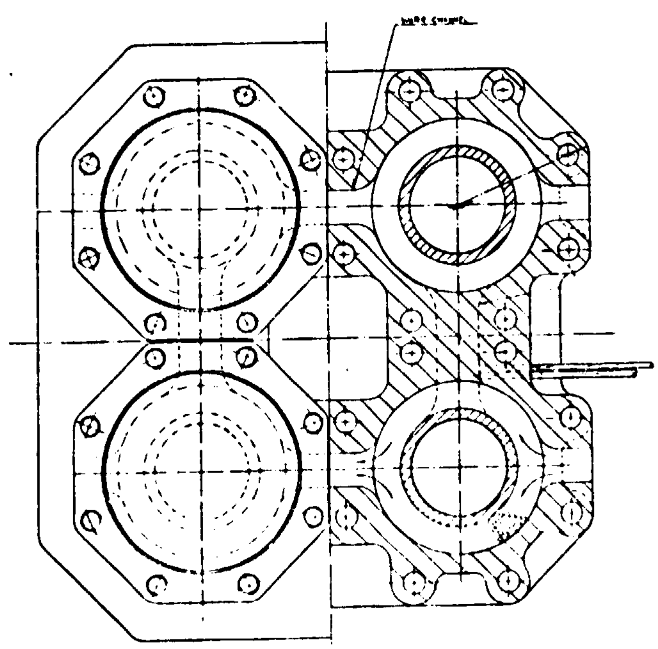


FIG 48: Top view, advanced kinematic Stirling engine

3.3.1 Heater design

3.3.1.1 General

The heater is the most critical component in the design of a solar powered Stirling engine.

The two types of heater design principles, indirect and direct heating offer both advantages and disadvantages.

The concepts for indirect heating meet the optimun cycle data, such as small dead volumes, uniform temperature distribution and will have optimum performance. However, the heat pipe is complex and includes high risk.

The direct heating concepts are simple but cannot meet the optimum data, which will decrease performance. Due to variable insolation flux distribution the temperature distribution over the heater cage area will not be uniform.

A number of heaters have been designed.

- one for indirect heating, including either a simple heat pipe device or General Electric's heat pipe with TES (fig 50-51)
- three different heaters for direct heating, annular regenerators (fig 53-57)
- two heaters for direct heating based on other design ideas. They will, however, not give optimum performance, but are enclosed to show the number of design possibilities giving high or low performance or high or low risk. The heater designs are made for one modified P40 engine (the location of the regenerators has been changed) and one 4 cylinder engine with regenerators surrounding the cylinders (fig 49, 58)

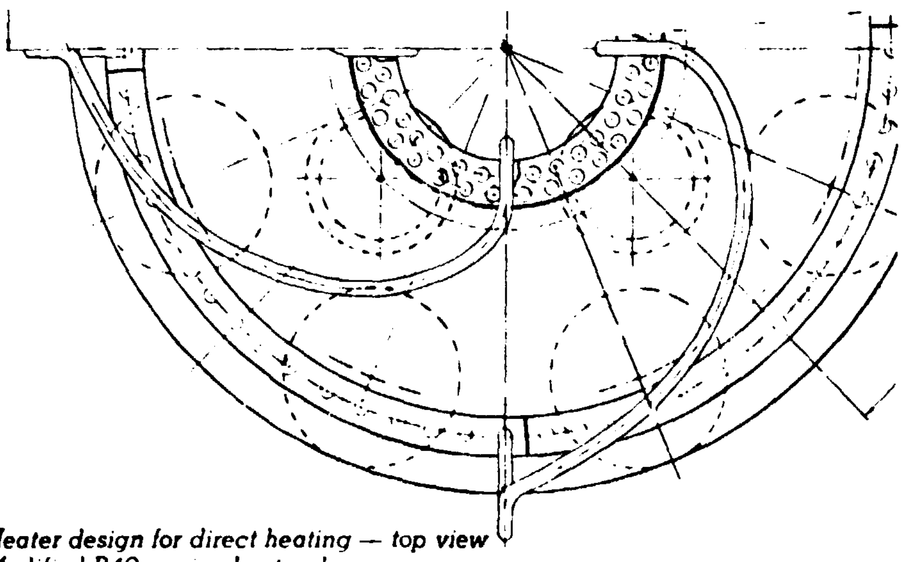


FIG 49: Heater design for direct heating — top view
Modified P40 engine heater design

3.3.1.2 Indirect heating

For indirect heating the heat will be supplied to the heater tubes by means of a heat pipe. The high heat transfer and no constraints in the design of tubes means that the tube shape can be governed by heat pipe requirements. The baseline heat pipe was the General Electric TES heat receiver shown in fig 38.

When designing the heater the data from the optimization can be fulfilled.

The cylinder/regenerator arrangement was designed with the only requirement that the engine must be adaptable to the GE heat pipe heat receiver. The interface between the cylinder/regenerator housing is a flexible corrugated plate (in two directions) to withstand thermal expansion (see fig 32). The location of the cylinder/regenerator housing will give room enough for the corrugated plate.

In addition a simple heat pipe heater head and heat receiver were conceptually designed. (fig 50 and 51).

The simple heat pipe has been divided into four separate quadrants, one per cylinder. The design will therefore not require any connections that will take up thermal expansion between the four cylinders.

The heater and heat pipe assembly will consist of four separate small heat pipes in connection with the cylinder/regenerator housing, less complex than the other design and easy to assemble and disassemble.

The tube design is a number of small tubes (sized for heat transfer on gas side) symmetrically fixed to the housing. The tube has heat transfer through the entire circumference.

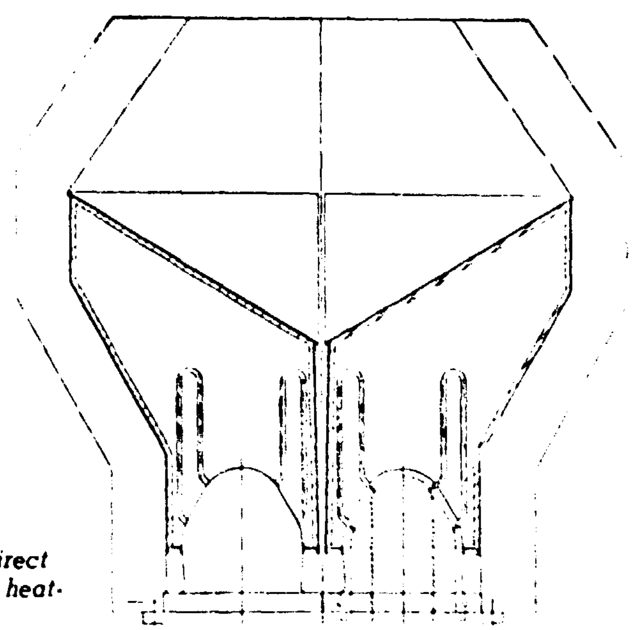


FIG 50: Heater design for indirect heating including the heat-pipe — cross-section

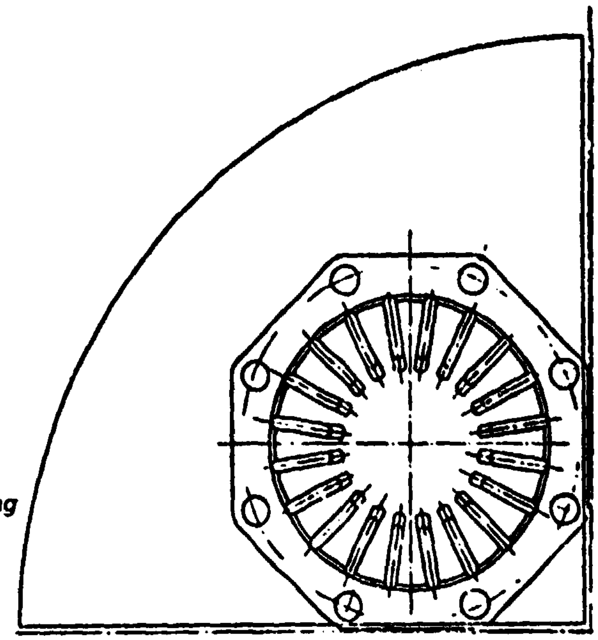


FIG 51: Heater for indirect heating — top view

3.3.1.3 Direct heating

The design of the heater for direct heating requires a more detailed analysis of the engine performance, including the heat flux distribution and limitations in the design.

A number of heaters for direct heating have been designed, based on different design philosophies. The design of the heater is not an isolated problem. The design of all the other Stirling components are affected by specific design parameters of the heater.

The different design philosophies for the directly heated heaters are:

1) An arrangement of gas passages (tubes or milled channels in a plate) that matches the requirements of the Stirling engine with receiver flux distribution, resulting in a uniform temperature distribution (see appendix 3, 4).

2) An arrangement of gas passages which in a uniform way, totally covers the heater cage area to exclude the use of surface extension or buffer material. The passages, consisting of tubes or milled channels in a plate will be located close to each other and only have a small clearance.

The different heater designs are shown in fig 53 - 57. Comments on the different heater designs will follow later in this paragraph.

The requirements mentioned in 1) above necessitate some form of buffer (or heat storage) material between the tubes (for example either a tube arrangement imbedded in copper or milled channels in a plate or a back surface which will reflect the incoming solar power on to the tubes or fin section).

If the heater cage diameter is increased, the heater tube length will increase and the design will depart from the optimum design.

The design of the heater of type 2 will, for small heater cage diameters, have a moderate tube length. For increased outer heater cage diameter (and also for decreased inner heater cage diameter) the tube length will increase very rapidly for this design philosophy, which will penalize performance. Also other limitations in design will set an upper limit for the outer heater cage diameter.

The cylinder/regenerator arrangement will play an important role in this heater design. The limitation in design (due to penalty on performance) are mainly due to the dead volumes (both on cylinder and regenerator side) and the unheated tube length, that will be a result of the design of the heater.

A parametric study of the heater was performed, which included the following items:

- total heater tube length
- unheated tube length
- dead volume between regenerator and heater
- dead volume between cylinder and heater

The results indicate:

- the tube length is not critical, rather long tubes, as for example, an increase of 50% will decrease efficiency~0.5% and power~1 kW. It depends somewhat on the distance from the optimization point.
- unheated tube length up to 25% and dead volumes up to 20 cm^3 at the regenerator will have a similar influence on performance.
- dead volume at the cylinder up to 20 cm^3 will have a negative influence on power~0.5 kW and cause only a slight decrease in efficiency.

Performance is shown in fig 52 for the different parameters.

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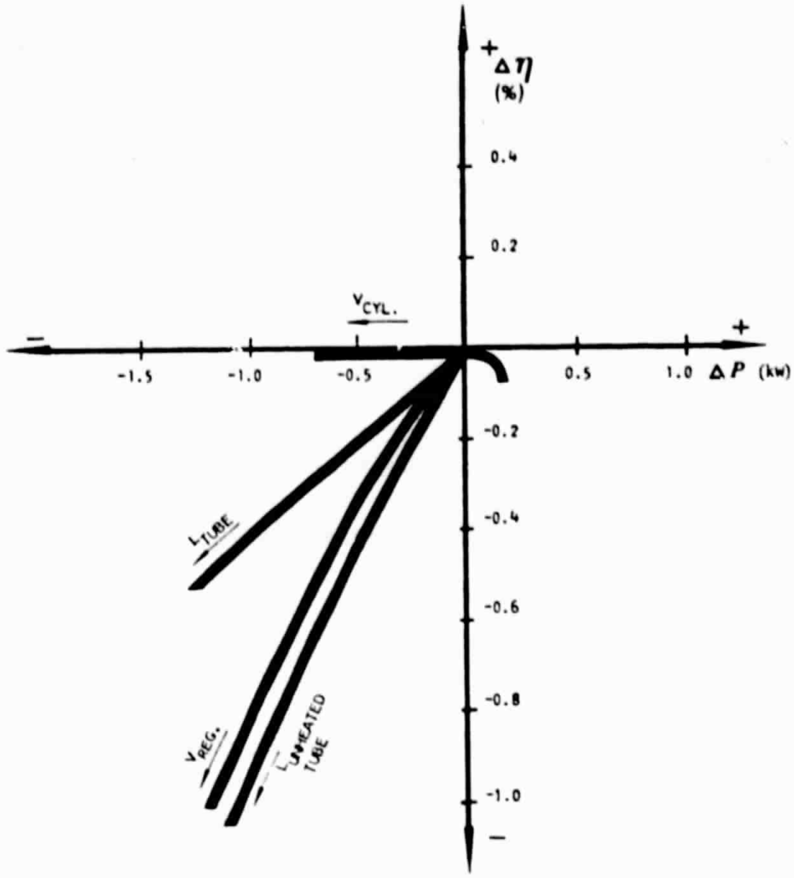

Fig 52. Influence on performance of engine parameters

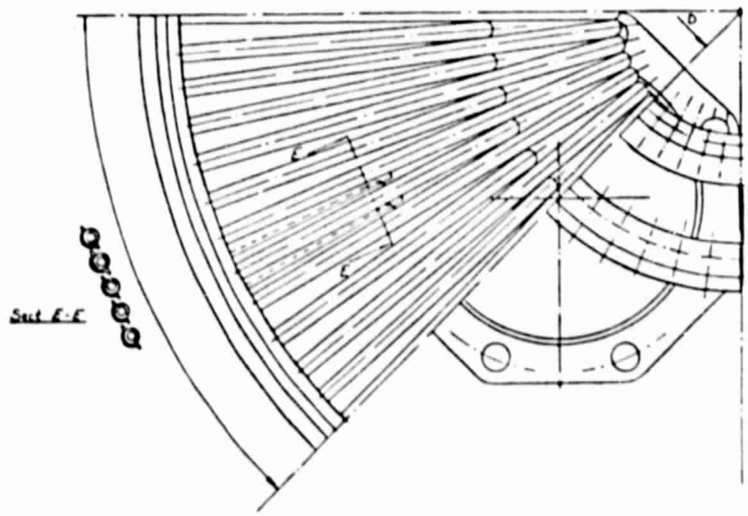

FIG 53: Heater design for direct heating - top view, straight tubes

Comments on the different heater designs for direct heating:

Based on the study of heat flux distribution in the receiver a cone angle of 60° has been chosen. (This figure has, however, not been optimized for this study). The selection was based on the JPL study of Appendix #4, which presented results for 45°, 60° and 75° cone angles. Section 3.3.2.1 further discusses this selection.

I) Figure 53, 54
Engine concept: 4 cyl, annular regenerators.
Flow passages: Straight tubes.

The tube design will be almost optimal for the engine. The tubes extend straight out from cylinder and regenerator manifold in radial direction to the outer heater cage diameter and connect to a duct of a small volume. The gap between the tubes will increase with the heater cage diameter. The heat flux in this area will be transferred to the tubes by means of fins brazed to the tubes.

II) Figure 54, 55
Engine concept: 4 cyl, annular regenerators.
Flow passages: Involute tubes.

This design incorporates involute tubes coverning most of the heater cage area. The tubes have not the involute form over all the length, and this will cause small gaps at some places in the heater cage area where heat is transferred via extended tube surface. The tubes are connected to a duct of a small volume at the outer diameter (same as of I). For involute tubes the tube length will increase compared to the optimum tube length, but there are advantages in better area coverage and greater tube flexibility for thermal expansion.

III) Figure 56, 57
Engine concept: 4 cyl, annular regenerators.
Flow passages: Milled passages in a plate.

This heater has many small passages, all of them having different lengths and different hydraulic diameters. The design is based on the requirement of keeping the same flow pressure drop and heat transfer for all tubes and to fulfil the requirement of total heater volume.

This heater design is optimum for the engine. The plate design, material stresses, etc, has not been thoroughly studied in this project. The design is based on a type of heater designed by Fairchild.

IV) Figure 49, 58
Engine concept: 4 cyl, 2 separate regenerators/cylinders.
Flow passages: Tubes.

The design is based on the existing P40 engine, with 4 cylinders and 2 separate regenerators per cylinder. The design incorporates the involute tubes covering most of the heater cage area. To obtain a wide outer diameter and at the same time have short tubes the inner diameter will be rather wide and the involute tubes span over an angle of 90'. The P40 design (67,5° angle) has to be modified and the regenerator location compared to the P40 engine has been changed.

V) Engine concept: 4 cyl 1 separate regenerator/cylinder.
Regenerators inside the cylinders.
Flow passages: Tubes.

An alternative engine design was made with regenerators surrounded by the cylinders. No drawings were made, however. The design incorporates the involute tubes covering most of the heater cage area. The lay-out of tubes is very close to the design (No. IV). The design was performed to show the different possible tube/cylinder/regenerator arrangement and the performance has been calculated to show the differences between the designs.

3.3.2 Heater performance

The different heater performances have been calculated and are compared in fig 59.

3.3.2.1 Temperature variation and drop in wall for direct solar heating

The temperature variation due to the uneven heat flux has been calculated. With some parts of the tubes receiving no heat input, due to design, the temperature variation will be 75°C between the maximum temperature and the minimum temperature (where the tube is unheated).

The variation caused by the actual heat flux distribution will be in the order of max 25°C, with heat flux distributions according to calculations performed by JPL (Appendix 4). The different distributions according to the cone angles will of course influence the performance. The influence is rather small and the difference between 60° and 75° cone angle is so small that optimization has not been performed. The results are based on 60° cone angle (fig 60).

The temperature drop through the heater tube wall has been calculated, the figure depending on the tube wall thickness. At the maximum it is 75°C. The tube dimensions differ in the calculations, but the final design for the directly heated heater has the tube dimension 4/7 .m (inner diameter/outer diameter). If small tubes are used, wall thickness also can be smaller and the temperature drop will be increased. The large (od/id) ratio was set because of life time requirement.

The variation in heat flux distribution around the tube will result in a temperature variation around the tube. This variation is indicated to be low, because the working fluid will aid the thermal distribution in the tube walls. However, we cannot calculate this distribution with our existing computer programs. We believe that the variation will not have a significant influence on performance.

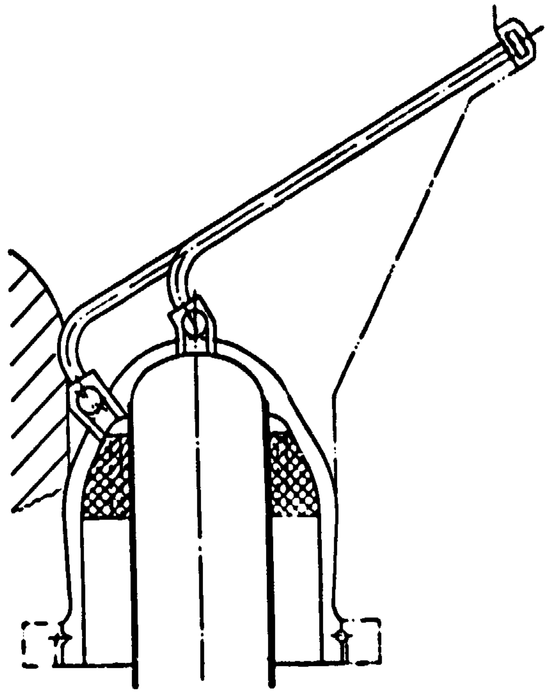

FIG 54: Heater design for direct heating - cross section

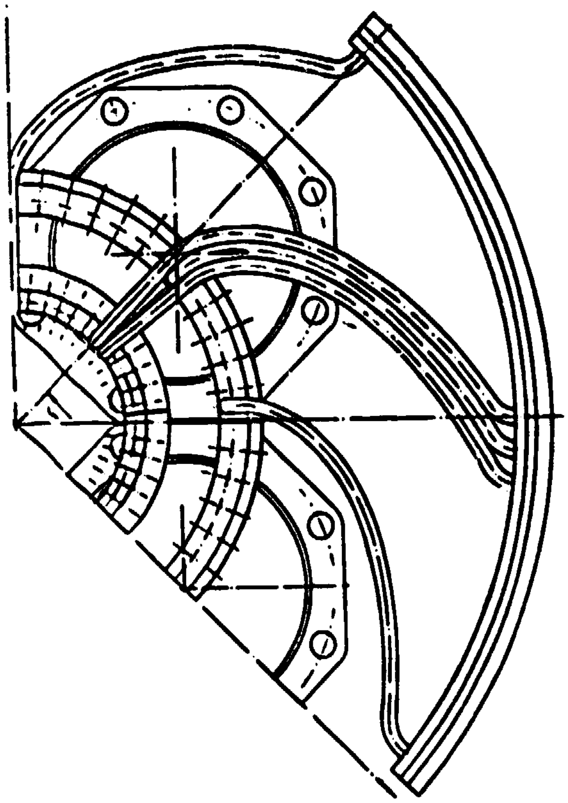

FIG 55: Heater design for direct heating - top view, involute tubes

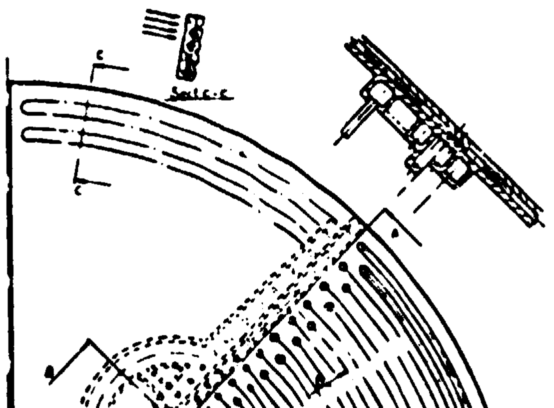

FIG 56: Heater design for direct heating - top view. Slots in a plate

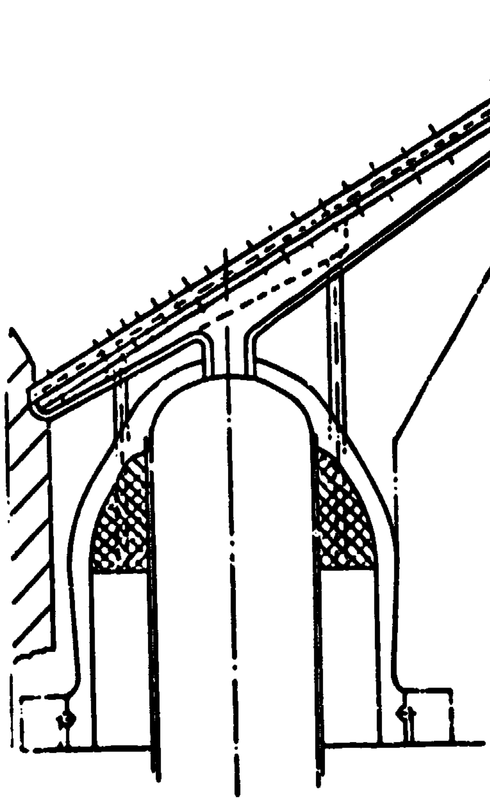

FIG 57: Heater design for direct heating - cross section

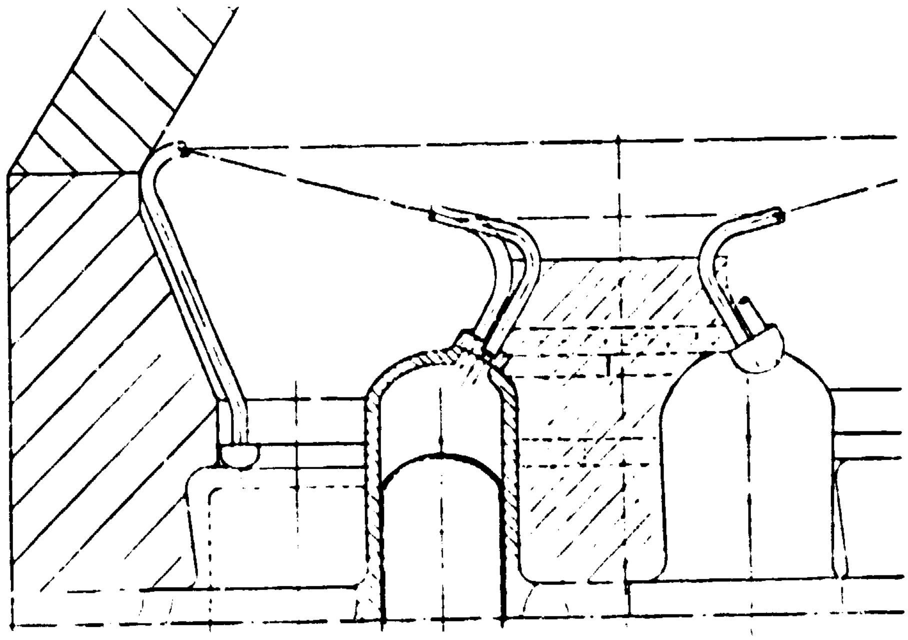

FIG 58: Heater design for direct heating · cross section

3.3.2.2 Power and efficiency

Figure 59 and 60 present the dimensions, parameters and calculated performance for 6 different heater concepts. The first is an indirect heated design which would require an intermediate heat transfer fluid, such as sodium in a heat pipe. The other 5 are direct solar heated designs of which 4 incorporate variations of tubular designs, while one involves a conical plate having machined channels in place of tubes. By the term "basic engine" is meant as one having the same dimensions of the thermodynamic components except for the heater.

The first four concepts all include annular regenerator, while the last two have more conventional separate regenerator housings.

For a fixed engine (swept volume, regenerator and coolor) the output power varies for the different heaters. This can be compensated, if the swept volume is somewhat increased. This will, however, change the design slightly to exactly fulfill the output power requirements. This iterative design step has not been performed. If the design is changed to fulfil output power the efficiencies will remain constant.

While there are considerable differences in heater quadrant dimensions, volumes and L/D ratios, the swept volume variation is only $\pm 5\%$ and the performance variations from fig 60 indicate a power variation of $\pm 1\frac{1}{2}$ kW ($\pm$ 6%) and efficiency variations within $\pm$ 2 percentage points ($\pm$ 4%).

For the same working medium the indirect heating concept shows the best performance. For direct heating use of hydrogen instead of helium causes an improvement of 2%-units in efficiency and 1 kW in power.

Fig 61 illustrates the effect of the relatively high internal heat transfer coefficient. In spite of large variations in heat flux, the temperature variations over the tube length are considerably moderated.

DIFFERENT HEATER CONCEPT COMPARED

(the heater designs are different, the basic* engine is, however, the same for all concepts)

	Heat Pipe Indirect Heated	DIRECT HEATED				
	Annular regenerator	Annular regenerator Tubular Heater		Annular regenerator plate heater	Modified P40 8 outer reg.	Four inner regenerators
Type heater	180° straight bent tubes	45° involute tubes	Straight tubes	Concentric channels	90° involute tubes	90° involute tubes
Cylinder to cyl distance (mm)	152	152	152	152	115	250
Pressure vessels (total)	4	4	4	4	12	8
Max dia heater (mm)	—	480	480	480	430	430
Number of tubes	12	12	12	26	15	15
Tube dia (inner) (mm)	4	4	4	3.25 max 0.8 min	4	4
Total tube length (mm)	280	560	420	—	520	450
Effect tube length (mm)	250	520	380	365 max 68 min	300	300
Tube effectiveness	0.90	0.93	0.90	0.78	0.58	0.67
Total length/dia	70	140	105	Average 130	130	112
Heater tubes flow area (cm^2)	1.5	1.5	1.5	1.6	1.9	1.9
Heater tubes volume (cm^3)	42	84	63	52 (40+12)	98	85
Dead volume at cylinder (cm^3)	0	12	12	19	10	24
Dead volume at regenerator (cm^3)	0	22	22	22	23.5	22
Dead volumes + heater volume (cm^3)	42	118	97	93	132	131
Piston swept volume	90	99	99	99	95	95

Per quadrant

* Basic engine is defined as one having the same dimensions of the thermodynamic components except for the heater. However, the engine configurations, such as cylinder spacing and regenerator shape may be different.

FIG 59: Comparisons of different heater designs

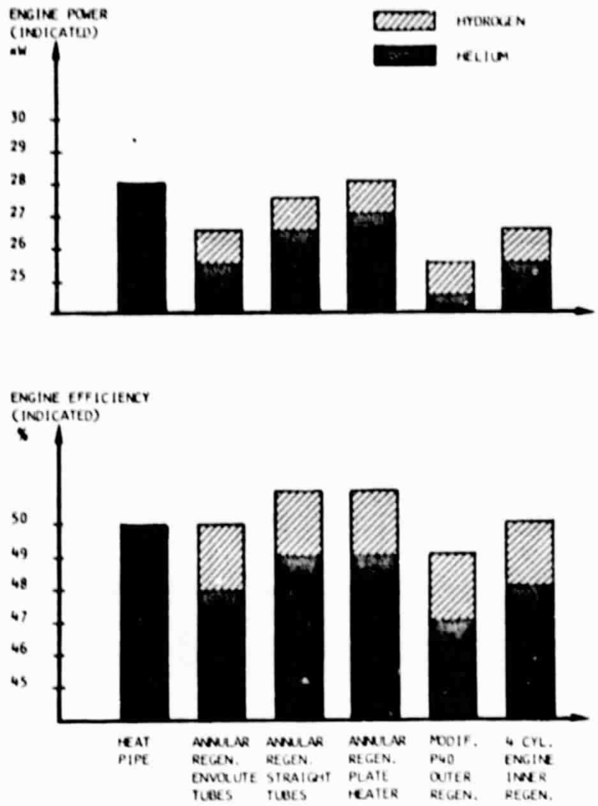

FIG 60: Engine performance for different heater concepts. Basic engine the same for all heaters

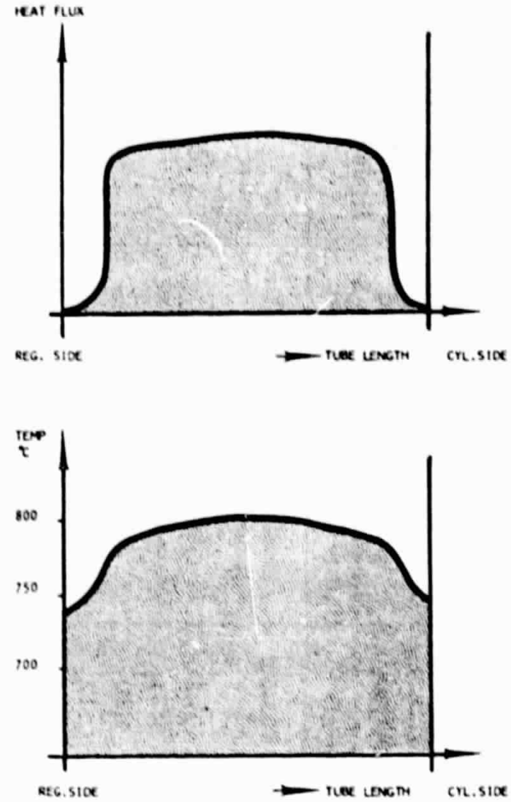

FIG 61: Calculated temperature distribution along a tube for a heater design for direct heating based on a typical heat-flux distribution

3.3.3 Cylinder/regenerator arrangement

3.3.3.1 Dead volumes

The lay-out of the cylinder/regenerator arrangement is very closely related to the de ign of the heater tube arrangement, but also related to the design of the drive system. For indirect heating a large number of housings will make the adaptation to the heat pipe more difficult, due to the bellows required for thermal expansion.

The requirements of the design are mainly:

1. Adapt the cylinder/regenerator arrangement to the tube arrangement as far as possible.
2. Minimize dead volumes both on regenerator side and cylinder side because the penantly on performance will be noticable.
3. Fulfil requirements of the drive system. (Center distances as needed for crank shafts, counter weights, balancing, bearings etc).

4. Provide enough regenerator area but avoid interference problems with packing the cylinder/regenerator housings together.
5. The distance between the cylinder/regenerator housings must allow sufficient room for bolts and nuts and flanges to mount the housings to the cylinder block/drive system.

3.3.3.2 Cylinder/regenerator housings

The design of the cylinder/regenerator housing with annular regenerator is a type of a cast housing with manifolds for the tubes. The separation of hot space and regenerator will be made by a thin partition wall in the cast housing. The cylinder walls (the sliding surface for the piston rings) will be either the partition wall or the cooler housing.

The housing is fixed to the engine body by using long bolts connecting the housings and the crank case. The regenerators have a tube form and surround the upper part of the cylinder. The regenerator design are either of cut wire* type or wrapped thin metal plates. The Stirling cooler is an annular type surrounding the cylinder. The cooler design is a housing containing a number of small tubes forming the gas passages for the Stirling cycle.

The cylinder/regenerator housing has been designed to meet the high life requirement of an engine for solar application.

The material used for the high temperature parts is Inconel 713. Fig 62 shows the performance of Inconel 713 at different operating condition.

*Thousands of short wires in random pattern in parallel planes.

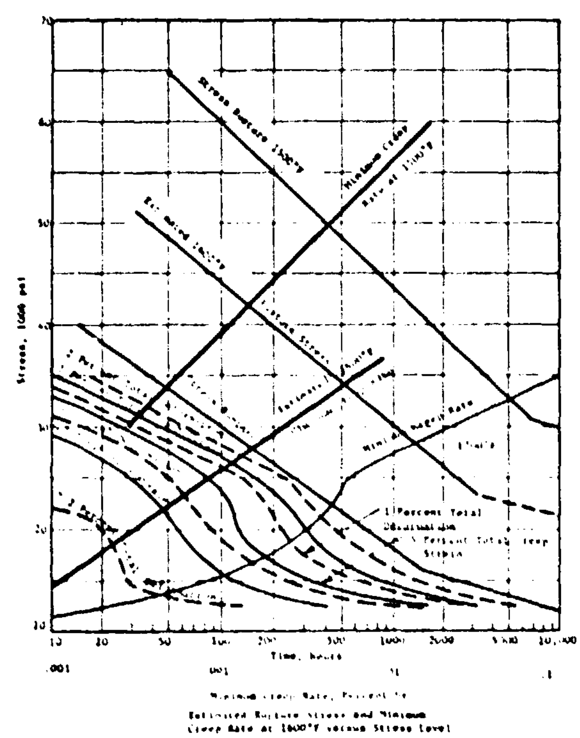

FIG 62: Material properties - Inconel 713 (from MTI solar engine study)

3.3.4 Cylinder block/seal system

The design principle of the engine body is to have the crank case and the cylinder regenerator housings mounted together by means of long bolts with the cylinder block in between.

The cylinder block includes the cold connecting ducts, the water cooling system and the seal housings with the seal systems.

The seal system consists of three parts; a sliding seal and a top seal with a diaphragm in between.

1. The main seal consists of an oil pumping polymer element, which prevents gas from leaking out from the cycle to the atmosphere and oil from entering the cycle. This polymer element maintains the gas pressure difference between the working gas and the atmosphere.

2. The top seal (cap seal) is a clearance seal which will smooth out the cycle pressure variations and give the volume below the cap seal a nearly constant pressure (max, min or mean depending on design of other components).

3. The diaphragm seal is a rubber diaphragm which will form a barrier between the working cycle and the volume below the sliding seal system to prevent oil from entering the cycle. The pressure drop over the diaphragm must be very low. The volumes over and below the diaphragm of each cylinder and the different cylinders are connected to each other, to get a constant pressure level. The diaphragm seal is redundant and serves as a back-up against oil leakage only, in case a small amount of oil should pass the main seal.

The cylinder block also includes the channels for supplying gas to the cycle and dumping gas from the cycle to a reservoir.

The supply of gas must, if the changes in power are rapid, be done when the working gas has its maximum pressure level to avoid output power drop. The supply of gas will be done through a recess in the piston rod, which only will supply gas when the cycle pressure is maximum.

For dumping the gas the flow passages will be in the cold region of the engine.

Different types of seal systems are shown in fig 63 - 66, which can be used as part of the seal system described above. If future testing of the PL seal (fig 66) proves as successful as those of the past year, it will most likely be recommended for all engine designs.

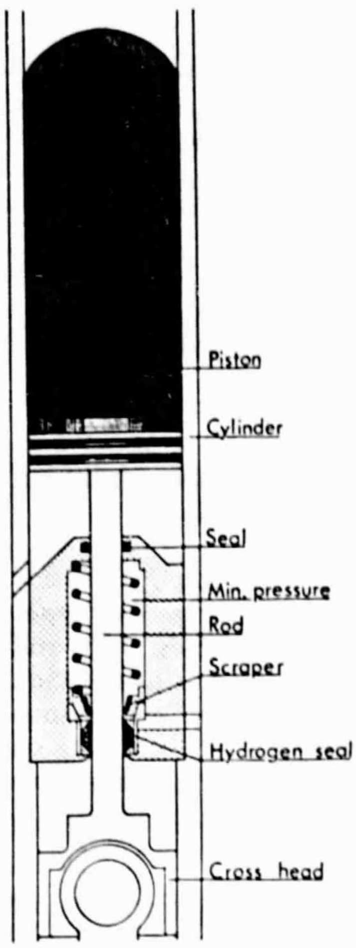

FIG 63: Sliding seal system

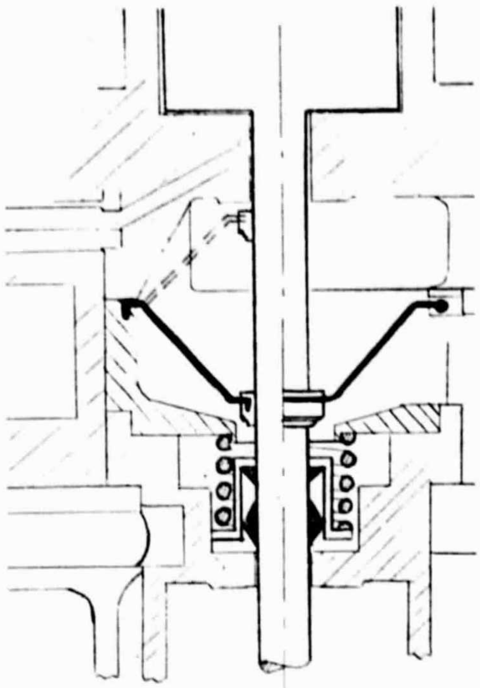

FIG 65: Diaphragm seal system

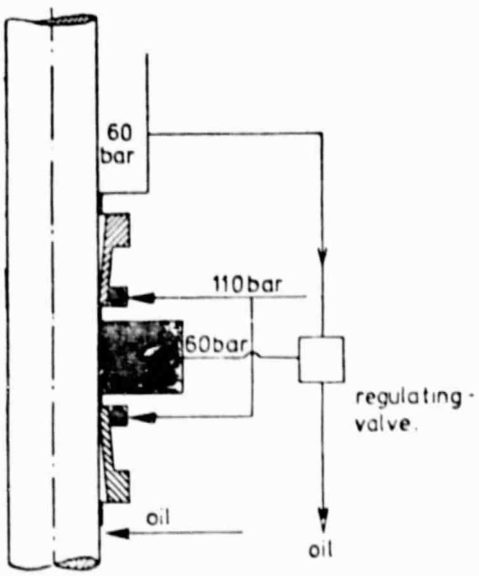

FIG 64: Dual pumping ring seal system

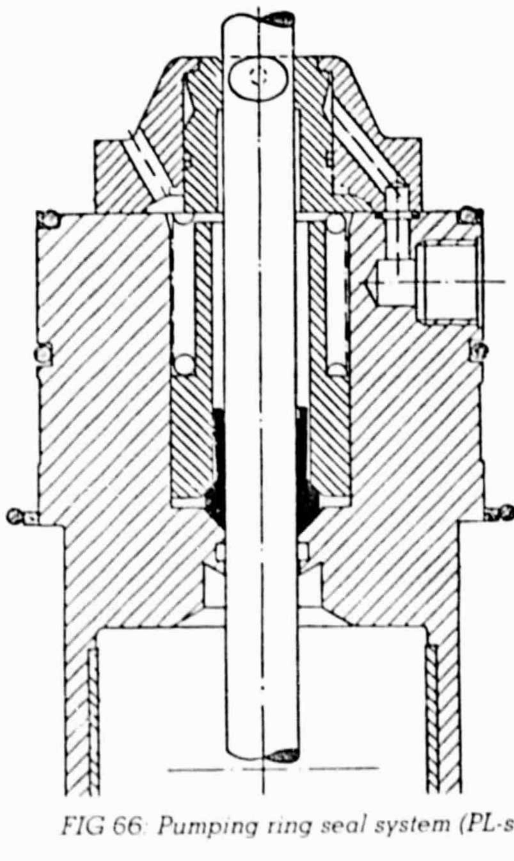

FIG 66: Pumping ring seal system (PL-seal)

3.3.5 Drive system and piston/piston rod assembly

The drive system design is based on the concept of two crank shafts and one output shaft, as used in the P40 engine. The configuration will give a compact, but rather heavy drive system and allows the use of parallel cylinders. However, unlike the P40 engine a link has been added between the slipper piston and the piston rod to minimize the motion of the piston rod relative to the main seal. The lubrication system must be specially designed for a solar engine. The design uses a separate oil tank. The different parts of the lubricating system have been placed in such a way that the oil will drain back to the tank at all engine elevations. Final design details of this system are not available.

Optional drive systems can be used in combination with the upper part of the engine. For example a swash plate drive system can be substituted for the U-drive system and create a unit shaped like a cylinder.

The seal system of the piston will separate the two Stirling cycles, one on upper part of piston, one on lower part of piston, acting. Two types of piston seals have been taken into consideration, the piston rings and the clearance seal (fig 67). To meet life requirements of this application the clearance shown in fig 67 may have to be used. Present experience with piston rings shows lower endurance. However, if maintenance on engine is allowed piston rings can be used. At present the piston rings are recommended until such time that sufficient testing of clearance seals is achieved.

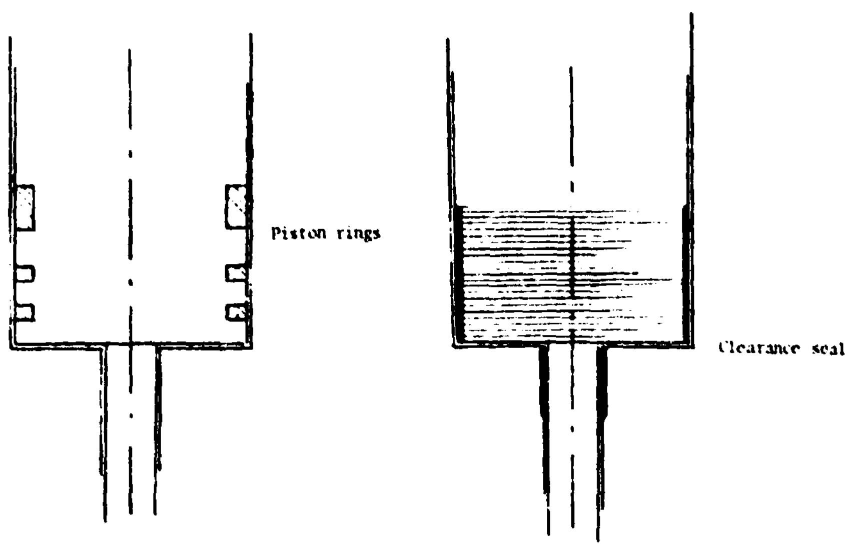

FIG 67: Piston seal systems

3.4 Stirling engine system and assumptions

The Stirling engine/(alternator) system includes

- the engine with heater head
- auxiliaries for selfsustained operation
- alternator
- control system

The auxiliaries/subsystem, which have been studied in this project are:

- power control compressor
- oil pump
- servo oil pump for power control
- electronic control unit
- gas control system

The weight of the system is shown in fig 68.

The main dimensions of the system are shown in fig 69.

The performance of the system is:

Output power: 20 kW el:

Conversion efficiency: 37-41 percent
(heat in to electric power out)

Assumptions:

- cooling water will be supplied to the engine with an input temperature of 43°C (110°F)
- electric power will be supplied to the electronic control unit
- start of engine will be made by the alternator
- gas to the engine is supplied from a gas bottle remote from the engine.

	Weight (kg)
Crank case (+ bedplate)	35.5
Crank shaft (X 2)	15.0
Drive shaft	2.0
Bearings	0.4
Gear wheels (X 3)	4.3
Connecting rods (X 4)	2.3
Cross head + pin (X 4)	1.8
Liner (X4) with plate	18.3
Seal housing (X 4)	6.8
Piston rod (X 4)	0.9
Piston + dome (X 4)	5.0
Cylinder (X 4)	9.2
Cyl/reg flange (X 4)	4.0
Heater tubes	2.4
Cylinder block	40.8
Regenerators (X 4)	3.6
Cooler (X 4)	9.5
Link + pin (X 4)	2.1
Lube oil pump assy with pinion	3.4
Oil filter + housing	1.0
Oil tank + oil	6.2
Servomotor (gassystem) power control	2.0
Gas compressor/control valve	9.0
Gas bottle servo valve	2.0
Water hoses + water	7.0
Bolts, tubes etc	11.0
Total	205.0
Alternator	120.0

FIG 68: Engine component weight

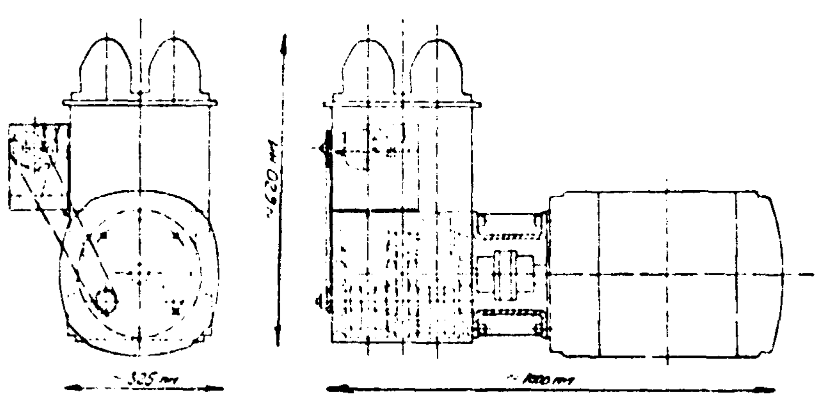

FIG 69: Engine/alternator outer dimensions

UNITED STIRLING

Summary results · Recommended for immediate application.

	Power (kW)	Efficiency (%)
Indicated cycle	27	50
Mechanical Friction	-5.4	80
Auxiliaries power consumption	-0.6	97
Alternator losses	-1.0	95
Total Systems	20	37

Operating conditions

Heater temperature:	800 °C
Coolant temperature:	43 °C
Mean pressure:	15 MPa
Working gas:	Helium

Advanced concept — 1985

	Power (kW)	Efficiency (%)
(Improvements)	—	3% units
Total	20	40%

		Direct Helium	Direct Hydrogen	Indirect Helium
Engine efficiency*	(%)	41	43	42
Alternator efficiency	(%)	95	95	95
System efficiency	(%)	39	41	40
Engine efficiency at low coolant temperature	(%)	44	46	45
System efficiency at low coolant temperature	(%)	42	44	43

* Engine efficiency includes mechanical friction, and auxiliary losses (heat in to shaft power out)

3.5 Summary results

Summary results of performance for a heat pipe engine including alternator and auxiliaries are:

Output power 20 kW_{el}

Conversion efficiency 37 %

The following table, also shown under 1.3.3, summarizes the specifications.

These figures are based on calculations with existing computer programs. These programs are valid for existing engine performance. We foresee, however, that engine development will continue and improvements of engine performance will take place until 1985. The improvements are of a magnitude corresponding to 3%-units, which will give the engine/alternator performance.

Outpower 20 kW_{el}

Conversion efficiency 40%

The improvements are basically to minimize the parasitic losses of the engine as for example

- friction losses (in bearings, gears, seals)
- flow losses (heater tubes, cooler, regenerator)
- distribution losses (heater, cooler)
- conduction losses (cylinder housings)

Further improvement of engine performance can be reached by decreasing cooling water temperature. The improvement is about 1%-unit per 10° C lower cooling water temperature.

The use of hydrogen instead of helium will improve performance. However, hydrogen cannot be used in combination with heat pipe and indirect heating. The improvement when changing from helium to hydrogen is about 2%-units. The comparison between the use of helium for an indirect heating type of heater and hydrogen for a direct heating type of heater will only give the net difference of 1%-unit (direct heating better than indirect), due to the more efficient heater design for indirect heating. This assumes that the comparison is made at moderate engine speed. At the upper speed limit the variations are much greater. It also assumes that the indirect heating version was optimized for He. If it was optimized for H_2, the efficiency would exceed the direct heated H_2 engine—but it is of only academic interest at the present stage of development.

Final results are shown on next page and in fig 70-72.

Bore	54 mm
Stroke	43 mm
Pressure ratio (P max / P min)	1.52
Number of cylinders	4
Swept volume per cylinder	99 cm^3
Number of heater tubes	12/cyl
Tube dimension	
(inner diameter/outer diameter)	4/6 mm
Tube length	258 mm
Number of cooler tubes	608/cyl
Tube dimension	
(inner diameter/outer diameter)	1/2 mm
Cooler length	65 mm
Regenerator type	Annular
Regenerator cross section area	4.860 mm^2/cyl
Regenerator length	70 mm
Filling factor	35%
Seal type	Pumping ring/diaphragm
Drive system type	U-engine two crankshafts
Alternator type	Induction (20 kWe output)
Input energy	54 kW
Losses — Rejected heat	26.5 kW
— Mechanical	5.5 kW
Engine efficiency*	41 %
Losses — auxiliaries	1 kW
Shaft power output (corr 20 kWe)	21 kW
Alternator losses (η 0.95)	1 kW
Engine and alternator combined efficiency	37 %
Output power — electrical	20 kWe
Engine speed	1.800 rpm
Working gas pressure	15 MPa
Working gas	Helium
Heater temperature	800° C
Coolant temperature	43° C

*Engine efficiency includes mechanical friction

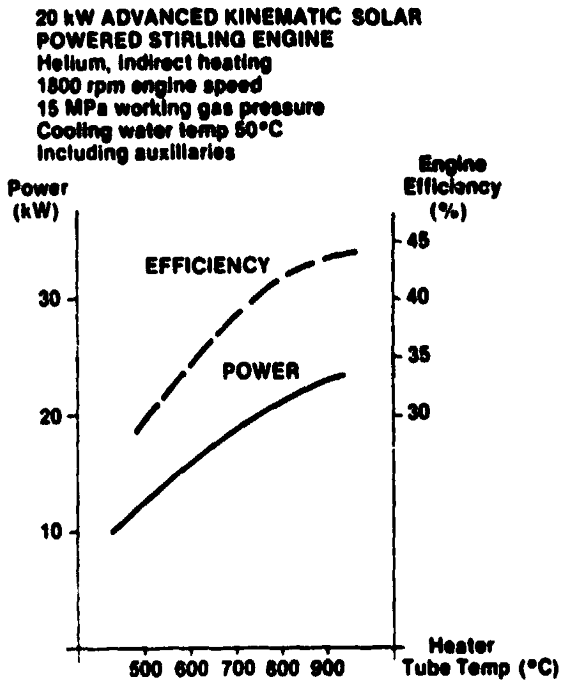

FIG 70: Final performance, advanced engine - power and efficiency at various engine speeds and working gas mean pressure (helium)

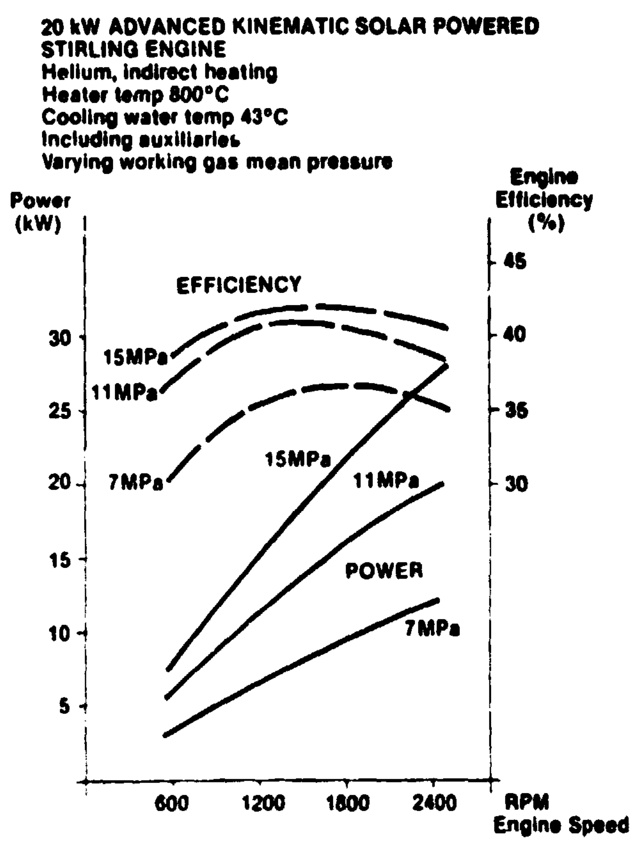

FIG 71: Final performance, advanced engine · power and efficiency at various heater temperatures.

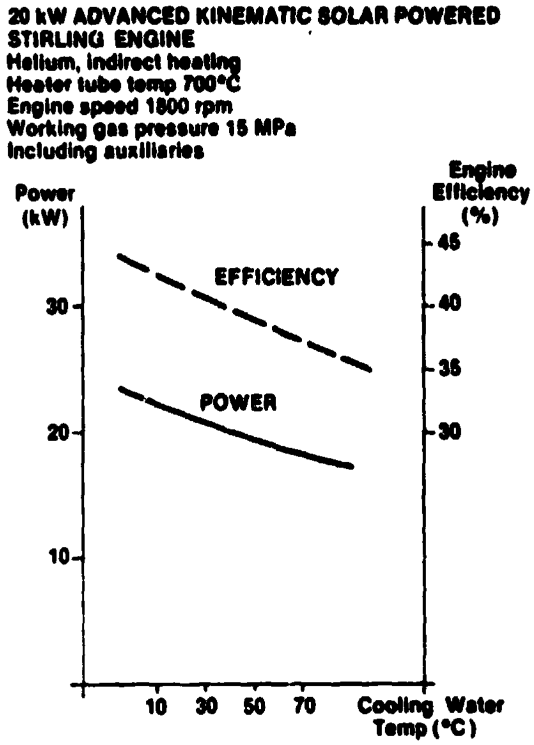

FIG 72: Final performance, advanced engine - power and efficiency at various coolant temperatures

3.6 Engine control

3.6.1 General

The control logic assumes that the engine is heated by solar insolation only, which means that the heat flux to the heater is given by the momentary solar intensity and cannot be controlled from the engine. To keep the temperature of the heater tubes constant the power output must be varied to match the insolation. This is done by controlling the working gas mean pressure.

In case the electrical grid cannot absorb all power produced, the power requirements can be met by means of shortcircuiting between all four cold spaces. This option requires, however, special design of the control system (presently used in automotive Stirling installations).

Small speed corrections to assist the alternator control in maintaining constant speed is possible if it is done with a control system with a sufficient short response time. This option requires also special design of the control system. To properly evaluate the influence of such a control mode on service, life, etc., of components such as seals requires a detailed specification of the load characteristics including demand for control of transients.

Start up, shut down and emergency modes for engine, insolation and load (grid), are described in the following sections. The control logic is shown in fig. 73.

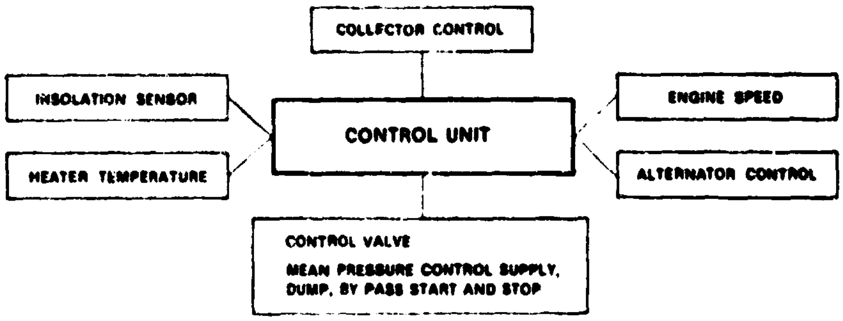

FIG 73: Control system

3.8.2 Start up mode

The collector is assumed to be kept defocused until engine is ready to start, to prevent overheating. The engine is cranked by the alternator which can be used a motor in the starting up mode.

A treshold value of insolation power must be attained before "starting" the engine (fig 74). At low insolation power, engine temperature will increase at a slow rate only. When the temperature of the heater tubes reaches 300-400° C the engine will begin rotating by its own force and the mean pressure may be increased. By motoring the engine, the temperatures in the heater will be smoothed out and the risk of overheating any tube will be minimum.

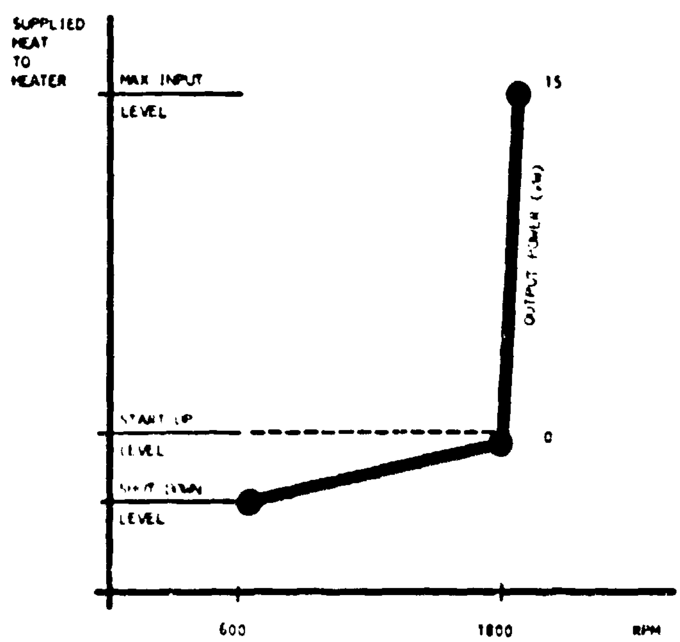

FIG 74: Control system performance · output power and corresponding input power

3.6.3 Shut down mode

If insolation decreases, the working gas pressure will decrease to keep the heater head temperature constant. This will be continued until, for a certain input power level (slightly below start up level), the output power goes to zero.

At this point the alternator has to be disconnected from the grid to prevent it from using power from the grid. There are then two possibilities

- either the engine/alternator is closed down and the collector defocused
- or the engine is turned into a stand by mode. In this case the engine rpm is controlled to match the insolation level and keep heater temperature constant.

The stand by mode can be maintained down to minimum idling rpm (600) at which the ultimate shut down level has been reached. Should insolation drop below this level defocusing of the collector becomes necessary and the engine will slowly come to a complete stop.

3.6.4 Emergency mode

If insolation power is maximum and the required power output from engine becomes zero, the system must be defocused. This will, however, take somewhere between 5 and 15 seconds.

During this time period the engine must be protected against overheating. This can be obtained by using the control mode by-pass, or shortcircuiting, which means that the four cold spaces are connected. Power output will be zero or negative depending on design, and heat will be absorbed by the heater and rejected from the cooler to the cooling water.

The heater tubes will, however, also act as a small buffer and will help to decrease response time for the control system.

3.6.5 Description of control system

Two types of control systems have been studied.

- mean pressure control system with control valve and servo unit (fig 75)
- mean pressure control system with supply and dump valves (fig 76)

Both the systems include a compressor which can be either engine-driven or separate, electrical-driven. The systems also include a gas storage (gas bottle).

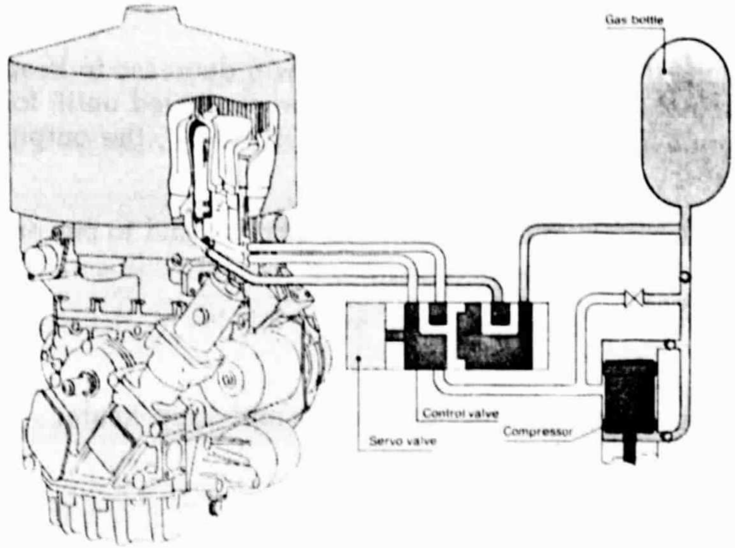

FIG 75: Power control system design - sliding valve system

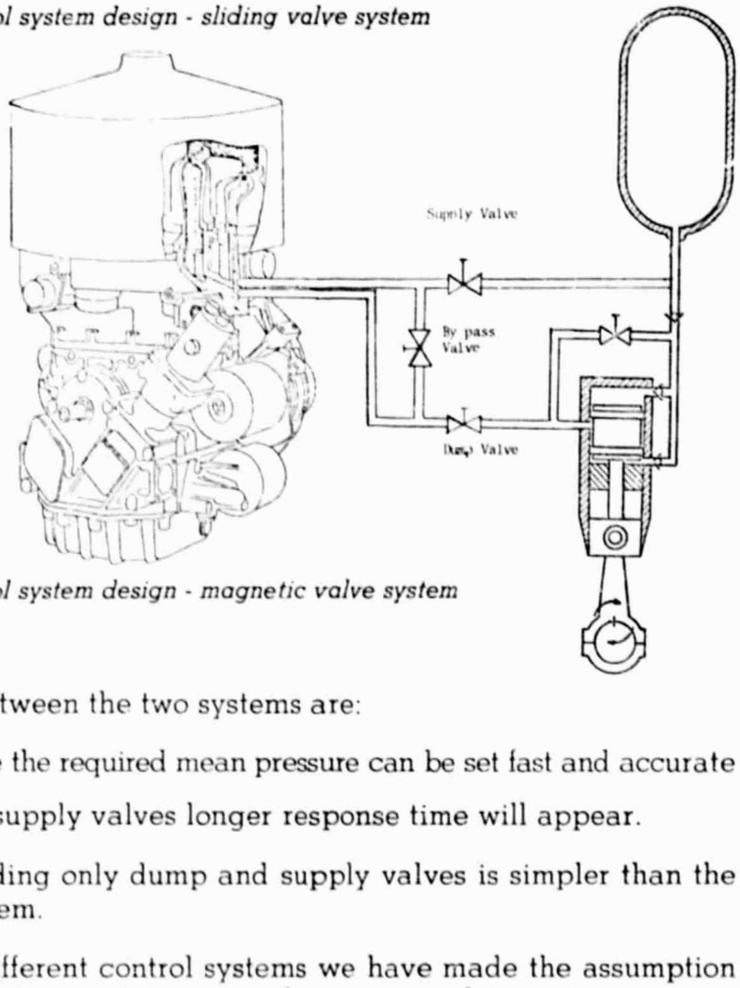

FIG 76: Power control system design - magnetic valve system

The difference between the two systems are:

- with control valve the required mean pressure can be set fast and accurate
- with dump and supply valves longer response time will appear.

The system including only dump and supply valves is simpler than the control valve system.

When studying different control systems we have made the assumption that the insolation time variations are slow compared to engine response time and also that some storage is included in the heating system. The effect of specific cloud cover on control dynamics is not known at this time.

Both the systems include safety systems (short-circuiting).

4 IMPLEMENTATION ASSESSMENT

4.1 Evaluation of state of the art and durability

4.1.1 Introduction

In this part of the ~~ ~~m the present status of the engine and components has been defined a... also the development activities, to meet the advanced engine requirements, have been defined.

The evaluation of present status of engine has been performed by evaluating component status of the prototype engines used at United Stirling. This evaluation shows that the engine performance will meet the current limited requirements, but further improvement in cost, life and reliability still remain. New technology for some components will be required which will influence performance as well as cost.

4.1.2 Component technology status

The 20 kW engine design development was reviewed in terms of those operating and design parameters which could significantly impact engine life, cost and performance. Parameters such as temperatures, stresses, clearances, complexity, materials, etc were evaluated for each engine component. As a result of this evaluation, those components which are considered critical to the engine operation were identified. The key technologies upon which the success or failure of each component part relies were also evaluated. The following definitions were adopted for the purpose of categorizing the status of each of the identified technologies.

1) Start-of-the-art

   This term refers to the current level of sophistication of a developing technology. A significant amount of experience has been accumulated with the technology as configured. The design and operational characteristics of this engine are well known.

2) Adaptation of current technology

   The technological concept has been proven in similar or analogous hardware systems. Adaptation to the solar Stirling design will require no basic research in materials or manufacturing.

3) Significant improvement in technology

   The concept is new or has only been verified in a laboratory. Some basic research is required in order to identify a proven design approach for adaptation to a Stirling engine system.

Component	Critical factors	Key technology	Technology Status
Heater head	Cost, life	Cast housings, brazed tubes	Improving present technology, new brazing techniques
Regenerator	Cost	Thin metal plates	New manufacturing technology
Cooler	Cost	Aluminium, dimpled tubes	New manufacture technology
Cylinder/piston pistondome/seal	Life	Close tolerance-seal or piston rings	New technology, improved material or coating technique
Pistonrod/seal	Life	Sliding seal	Improving present technology
Cylinderblock/drive	—	Cast block, U-drive	Improving present technology
Power control	Cost, life	Mean pressure control—sliding valve, or magnetic valves	Improving present technology
Control system	Cost	Electronic system (micro computer)	Improving present technology

FIG 77: Component critical factors.

Component	Technology	Cost	Life	Maintenance	Reliability
Heater head	Cast housings Brazed tubes	Lower manuf. cost	Simple design increases life	Creep stress gives limited life. (Also fatigue due to start and stop)	Simple design—less susceptible to failure
Regenerator	Cut wire or thin plate	Lower material cost	—	—	—
Cooler	Aluminium, tubes with dimples	Lower material cost	—	—	—
Cyl/piston piston seal	Piston rings or close tolerance seal*	*Precision Machinery (high cost)	Limited wear	Low wear comp.	Slight performance decrease due to wear
Piston and/ seal system	Sliding seal and diaphragm	—	Limited wear	Low wear comp.	Slight performance decrease due to wear
Cylinder block	Cast block or water jacket	—	—	—	—
Drive system	U-drive	—	—	—	—
Power control	Mean pressure control valves	A number of comp. included	Life of valves limited	Low wear comp.	Slight performance decrease due to wear of check valves
Control system	Electronic Control system	Low cost Microcomputer	—	—	—

FIG 78: Component critical factors

4.2 Production cost

A production cost evaluation has been performed by a consultant, thoroughly familiar with production cost for Otto and diesel engines. The consultant has earlier been involved in the evaluation of the cost for other types of Stirling engines. The results have been followed up at United Stirling by the redesign of some engine components having high cost.

Presented graphs show

I) the variation of production cost for different number of units (fig 79)

II) comparison of direct labour and material cost (fig 80)

III) cost for engine, receiver and alternator respectively (fig 81)

The production cost is based on an arbitrary production rate of 100 000 units per year. This appears to be the right order of magnitude if solar thermal goals suggested by the Federal Government are to be met by the year 2000.

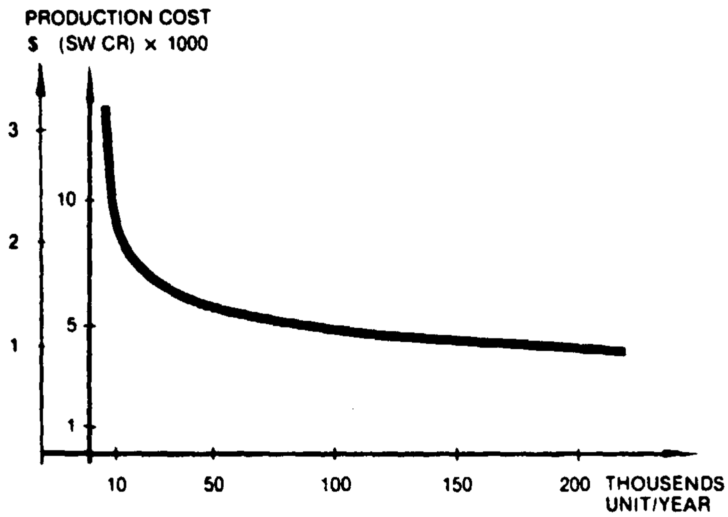

FIG 79: Production cost at various production rates

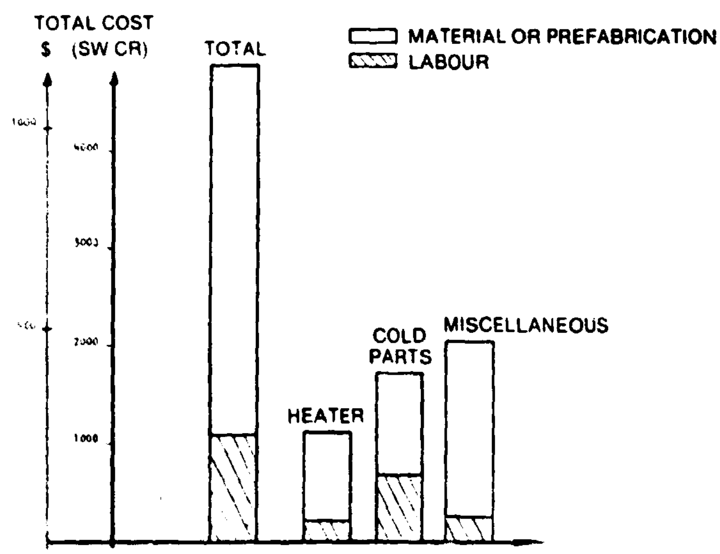

FIG 80: Comparison of cost - labour and material

COST—ENGINE/ALTERNATOR SYSTEM

	Sw cr	$ *
ENGINE	4900:-	1166
ALTERNATOR	3000:-	714
HEAT PIPE DEVICE	2500:-	595
TOTAL	10400:-	2475

* 1979 Conversion rate $ 0.238/Sw cr

FIG 81: Total system cost

4.2.1 Summary results

Production cost of the engine including auxiliaries (excl. alternator) is about 1 200 $ per unit (figure based on Swedish experience). Comparison shows a 10 - 20% lower figure based on US experience. The heater is the most expensive component — approximately 22% of the total cost.

The total cost for engine, alternator and heat pipe will be about 2 500 $.

4.3 Scaling of engines

4.3.1

General installation requirements for upscaled engines are not defined. These requirements will very much influence the engine design, for example if the engine shall be ground mounted or fixed to a collector, tracking the sun. No special requirements have been taken into consideration and the engine designs discussed later on are based on common design principles of Stirling engines.

The power level of Stirling engines can be increased in several ways.

1) Increased swept volume

2) Increased rpm, pressure and/or temperature.

Increased swept volume will require an upscaled design.

Increased rpm, pressure and/or temperature will retain the engine design - however, some critical components as cylinder/regenerator housings, piston, heater tubes, etc have to be redesigned. An increase of either the pressure or the temperature will require new materials for the hot side components - especially the heater.

4.3.2 Engine design

Engine scale-up has been done by assuming geometric similarity and applying the conventional engine scaling rules where γ is defined as the linear scale factor and

Power $\sim \gamma^2$

RPM $\sim 1/\gamma$

Weight/power $\sim \gamma$

Volume/power $\sim \gamma$

Engine working fluid pressures and temperatures are assumed to remain constant for the scaled engine and efficiency remains nearly constant. (More detailed information of scaling-up engines, see ref 1.)

Scaling rules (reference IECEC paper 789355) show that if the power increases the rpm will decrease.

However, if the scaling factor is small the rpm can be kept constant and only the swept volume will increase (cross section areas will increase, lengths will remain constant). The limit of using the present advanced kinematic solar engine will be up to power levels of about 40 kW electrical output. The diameter of the cylinder/regenerator housings will set this limit due to among other things, housing stresses and drive center to center distance (fig 82 and 83).

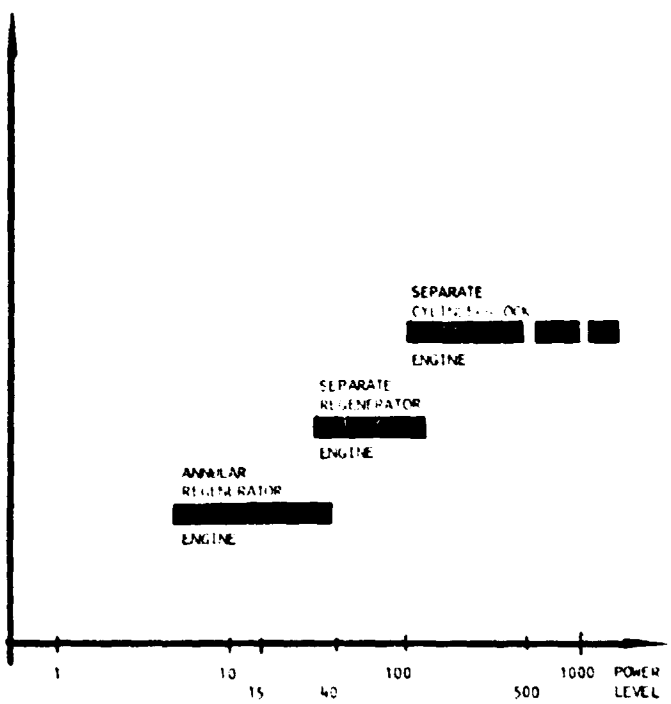

FIG 82: Engine concepts at various power levels

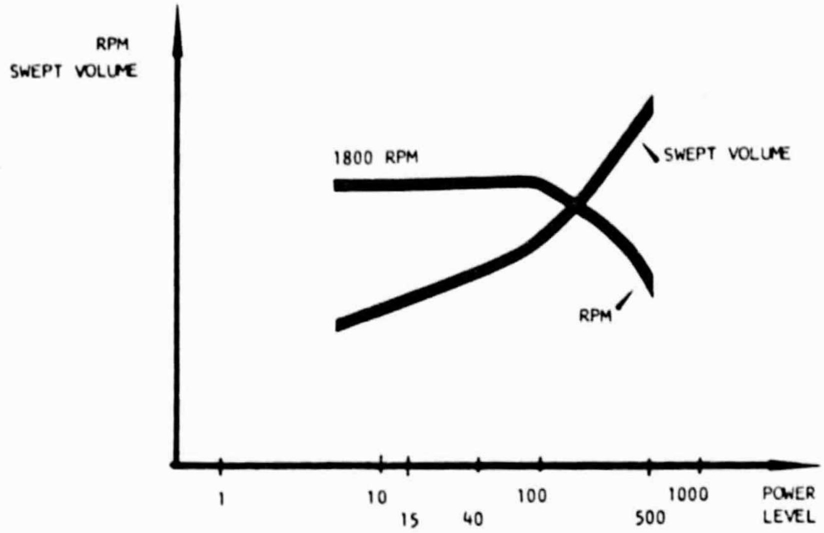

FIG 83: Engine design parameters - rpm and swept volume - at various power levels

If the power level is further increased, separate cylinders and regenerators have to be used. A new heater configuration has also to be designed if separate housings will be used. This type of engine configuration is suitable for output levels up to 100 kW. The engine speed will be held constant, 1 800 rpm for this design to get 60 Hz output. No special advantages will arise if engine design speed will be decreased.

For higher power levels - above 100 kW - the compact engine design - all cylinders and regenerators together in one cylinder block - will be difficult to use.

Instead the engine should use a common drive unit but let each cylinder and its regenerators comprise a separate heater head. This design gives a flexible engine using the efficient double-acting principle. The single drive system allows use of a simple interface to the electrical generating system. Problems arise, however, to get the heat input to the engine in a simple and effective way, but this has not been worked on.

Of course, electrical generation can be performed by connecting a number of small units. This design will, however, be complex and a thorough analysis of the transmission system has to be performed.

4.3.3 Performance

For larger engines the flow losses will increase if speed is constant and due to this, the engine will be designed for lower rpm.

The efficiency of double-acting engines will increase slightly for higher power levels due to relatively lower parasitic losses (fig 84). Otherwise no special improvements of the Stirling thermodynamic cycle can be expected when increasing the power output of the engine. However, piston ring and sliding seal leakage should decrease with engine size, for a constant clearance.

The evaluation of the different engine designs will not imply any significant increases in performance beyond what is mentioned before.

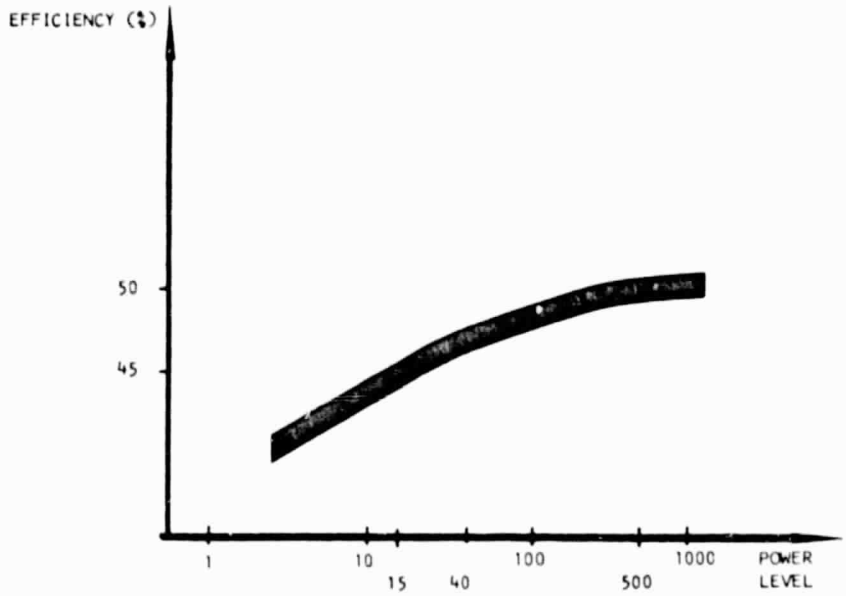

FIG 84: Engine performance - efficiency - at various power levels

4.3.4 Risk

The critical factor when scaling up the engine to higher power will be the design of the engine heater head. This has to be taken into consideration very carefully when discussing high output power levels and connecting either a number of small units on the electrical side or scaling up the engine/alternator to one single big unit.

STIRLING ENGINE HEAT PIPE SYSTEMS

Stirling engine characteristics

The efficiency of the Stirling engine is strongly dependent on

- the temperature level at which energy is transferred to the working medium

- speed of rotation

- choice of working medium

The efficiency for a given engine configuration tends to drop at high operational speed. In an engine with requirement of a high specific power the working medium should be helium or preferably hydrogen, which results in an increase of specific power at constant efficiency.

Inherently the Stirling engine process calls for high temperatures and high pressures of the working medium to yield a high efficiency.

The speed limitation of the Stirling engine is mainly to be found in the design of its heater head, particularly in the case of designs for fossil fuel heating. Due to the relatively low heat transfer from the combustion gas to the heater tube walls a fairly large surface area is needed. Optimization of the heat transfer on the outside tends to increase the heater tube diameters. However, considering the inside of the heater tube where the working medium absorbs the energy, such an increase in diameter is detrimental to the heat transfer. The heater assembly therefore must be a compromise, a compromise which becomes more severe as the specific power of the engine is increased.

By means of a heat pipe system large amount of energy may be transported to small surfaces with very low temperature differences. Using the indirect principle of the heat transfer the Stirling cycle may be optimized without regard to the previously limiting factors. Thus, short heater tubes of smaller diameters may be used, making possible a design with considerable lower pressure loss in the working gas and thereby allowing an increase of the rotational speed and hence a decrease in the weight to power ratio of the engine.

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The heat pipe

The heat pipe for high temperatures (500 - 1000° C) consists of an evacuated vessel, a porous wick and a liquid metal as a working fluid. To ensure reliable performance the design and selection of material must be made with consideration of

- Mechanical stability
- Chemical stability to the surrounding environment
- Liquid transport
- Purity of the working fluid
- Desired life time

Liquids, most suitable for heat pipe operation for thermodynamic reasons, are potassium in the 400-800° C temperature range and sodium in the 500-900°C temperature region. Approximation theories for dimensioning of the heat pipe are given by Cotter & Busse. These models give rather realistic values which, to some extent, may be improved by using experimentally verified values of parameters appearing in the respective model.

General design

The elements involved in the design of the heat pipe are

- Container tube and condensing area
- Wick structure

Stress analysis and discussions of material compatibility for the containers are given very close considerations. One of the basic requirements of any selfcontained vapour heat transfer system is a means of returning the condensed liquid to the evaporator to replenish the supply. In any system chosen this return may be accomplished either by gravity, or by pump, or as in a heat pipe by means of the capillary action. The criteria for selecting a heat pipe wick structure include resistance to working fluid circulation, capillary pumping capacity and evaporated heat flux capability.

Optimization calculations of the heat pipe may be performed taken into consideration

- Pressure drop both in the vapour and the liquid phase
- Temperature drop

Design

Caiculation, design, manufacturing and testing of a Stirling engine heat pipe system have been carried out at United Stirling. It was performed as an experiment with the objective to demonstrate the possibility of using a sodium heat pipe in conjunction with a Stirling engine.

The total heat convection loop in this experiment consisted of

- A Stirling engine functioning as a condensor in the heat pipe
- The sodium heat pipe
- An electrical resistance heater functioning as the heat source

Engine design

The engine used in this experiment was a single cylinder displacement type engine. The difference in the engine for the heat pipe application was in the design of the heater head. The heater tube diameter was reduced, smooth radii were introduced to decrease pressure drop and thereby allow increasing rotational speed. The dimensions of the heater head were optimized for maximum power by means of highly developed computer programs. The heater head configurations consist of a number of tubes on to which the sodium vapour condensates and yields its heat of evaporation. The cooler regenerator units were kept unchanged from the conventional engine design.

Heat pipe design

The evaporator section of the heat pipe was chosen of two different configurations. (Fig 1). One type was designed and manufactured from a stainless steel tube of outer diameter $d_o = 121$ mm and wall thickness $t = 3.5$ mm. The total length of the tubes was 450 mm. The heat pipe was connected to the Stirling engine via a conical shaped adiabatic section to a dome.

This dome was in turn welded to a plate that was brazed to the cylinder and regenerator cups and thereby makes up a completely sealed vessel.

The condensor section was as mentioned previously made up by the Stirling engine heater tubes. The condensed sodium on the heater tubes will flow on to a cover plate on to which was applied stainless steel gauze layers. From this layer the liquid sodium was transported back to the evaporated section by means of the capillary structure made up by a number of layers of gauze.

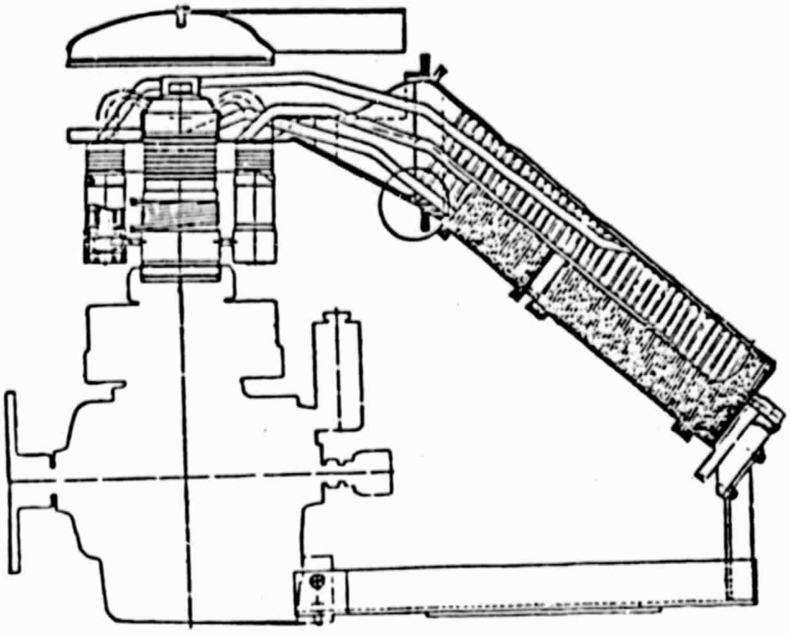

FIG 1: Principal design

The heat supply to the evaporator section was arranged by electrical resistance wires based in grooves at the outside of the tube to assure perfect contact between the element and the evaporator wall. (Fig 2).

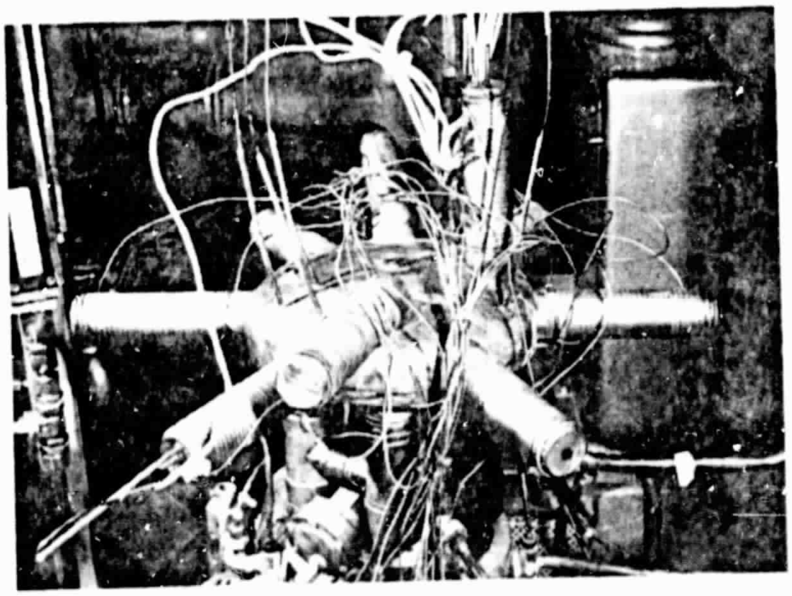

FIG 2: Test equipment

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Experimental apparatus

The test object was installed in a test cell and connected to a Schenk U 16 water brake. The shaft power output of the engine was controlled by varying the mean pressure in the cycle. The brake was fitted with a D/C motor for distant control of the torque. All control functions were mounted on a panel in the manoeuvre room. Safety precautions were taken to prevent water and sodium contact in case of accidential leakage from the heat pipe.

The electrical input was controlled for maximum power input by means of tyristors.

All temperatures were measured with Chromel-Alumel thermocouples. The thermocouples were calibrated at 600°C to a relative error leading lower than 0.25°C between the different thermocouples. Temperatures were measured on the outside of the heat pipe vessel, in the liquid as well as the vapor phase in several different locations and were recorded continuously. A total amount of approximately 180 hours of engine operation were clocked. The general results showed that Stirling engine with a high temperature sodium heat pipe resulted in an appreciable increase in specific power and provided valuable information for the engineers for further works in this area (Fig 3).

SINGLE ACTING ENGINE 1:98
MEAN PRESSURE 11.5MPa
COOLING WATER TEMP 15°C
INDIRECT HEATING HEATPIPE He ------
CONV. HEATING H_2
CONV. HEATING He

COMPARISON OF INDIRECT HEATING (HEATPIPE) WITH CONVENTIONAL HEATING. SHAFT POWER VERSUS ROTATIONAL SPEED

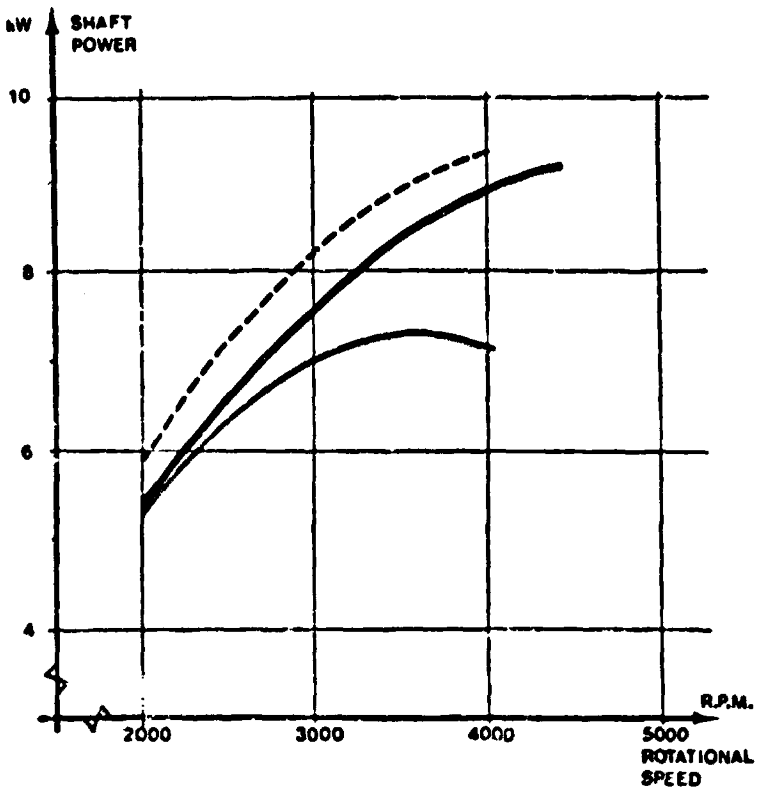


FIG 3: Engine performance

DISCUSSIONS OF BEYOND 1985. ENGINE CONCEPT

Ceramics

Introduction

United Stirling has for some years been interested in ceramics for use in the hot parts of the engine. Ceramics are interesting because of their special high temperature properties, as for example:

- for some materials high material strength which is retained up to 1 200 - 1 600°C

- some materials, especially glass ceramics, have low heat expansion which will cause low and intermediate thermal stresses.

Designs for higher working temperatures are possible and the heater working temperature can be increased to 1 000 - 1 100°C, if ceramics are used.

Materials

Ceramics keep their material strength up to high temperatures. Heat expansions are varying from 0 for some glass ceramics up to 30% of metallic high temperature alloys for silicon carbide. The material heat transfer capability is a property we can derive advantage from. Some ceramics show a much greater heat conductivity than those metallic materials we now use and this can be used in designs where heat is transferred through a wall. Silicon carbide can be used in the heater design for this reason. On the other hand some ceramics show a much lower heat conductivity than metallic materials and can be used in designs where we want as low heat conduction losses as possible. Silicon nitride can therefore be used in cylinder and regenerator housings where as low heat losses into the cooling water as possible are wanted.

Further advantages of ceramics are:

- phase stability at high temperatures

- corrosion resistance

- oxidation resistance

Adequate material tests have not been performed. Tests have usually been performed in oxidizing but not in reducing atmosphere. Some types of materials will need preoxidizing.

The densities of ceramics are low; silicon carbide and silicon nitride both have almost the same density as aluminium. A low total engine weight will be an advantage if ceramics are used.

Availability of raw materials is good. The disadvantage of ceramics from a design point of view is their brittleness. Strong forces between the metallic and non-metallic atoms give the ceramics high resistance against deformation, quite different from metallic materials. Concentrations of high stress in pores, microcracks and other inhomogenities cannot be smoothed out through plastic deformations. The material strength will depend much on the material structure, and quality requirements for raw materials will be high. The material strength will depend on how pure the material can be made and also how it will be formed and sintered.

The brittleness of the ceramics will in practice cause a wide spread of measured values of the material strength between different samples of the same material both due to dimension relations - larger test samples have lower mean material strength than smaller test samples - and to stochastic variations from one sample to the other.

The manufacturer must guarantee right properties and constant quality of his ceramic material. He will have to perform non-destructive tests between different stages in the manufacturing process. Such non-destructive tests methods are not available to any extent and need to be developed. Some type of agreement between manufacturer and customer regarding material characterization is needed to establish a standard for test methods. Finally a new design philosophy is required. It will take considerable time and effort before we reach a level of knowledge allowing ceramic components design to be as reliable as metallic components design.

The types of ceramic materials which probably are most interesting for use in a Stirling engine are SIC (silicon carbide), Si_3N_4 (silicon nitride) and SIALON (Si-Al-O-N).

Excerpt of

Optical system of the advanced dish
- Stirling module.
P Poon, S Higgins, JPL.

Calculation of receiver heat-flux distribution

OPTICAL SYSTEM OF THE ADVANCED DISH-STIRLING MODULE

Peter Póon and Sandra Higgins
Jet Propulsion Laboratory

Introduction

The optical system consists of the paraboloidal reflector of diameter 10 m with the center of the receiver aperture located at the focus. The rim angle is 45° and the F/D ratio is 0.603. Ideally the surface of the solar concentrator should correspond exactly to that determined precisely by the geometric relationship, and the optical axis of the paraboloid should be oriented along the direction of the sun. In practice, there are various errors that will degrade the optical performance from the ideal situation. These errors may be divided into three categories:

- Pointing or tracking errors
- Surface slope errors
- Specular angular spreading

Tracking Error

The trackering or pointing error is the angular offset of the direction of the solar beam from the center of the receiver. It arises from any sensor misalignments, the solar tracker control offsets and hysteresis, and receiver support deflection caused by gravity or wind load as the concentrator changes its orientation while tracking the sun. The net effect of tracking errors, provided that they are small, is to shift the flux distributions at the focal plane with very little distortion in their overall profile. The specification for the tracking mechanism and control design of the ADS# 1 is that the tracking error will not exceed 1.7 mrad (0.1°) for normal operating conditions in steady state winds of 50 Km ph. Quantitatively the tracking error of 1.7 mrad causes a shift of the flux distribution at the focal plane by approximately 1 cm. The size of the receiver aperture must be enlarged to accomodate the shift. Furthermore, the control system transient response will be such that the tracking error will return to within 1.7 mrad in less than 20 seconds from the onset of a 20 percent gust condition.

Surface Slope Error

The surface slope error is the angular deviation of the actual surface normal of the fabricated concentrator from that of the ideal geometric surface, and is most instrumental in spreading the flux distribution at the focus. It results from a number of sources, for example, macro-roughness due to manufacturing methods, subassembly manufacturing errors, installation misalignments, and distortions as well as structural deflections due to external forces. A statistical..

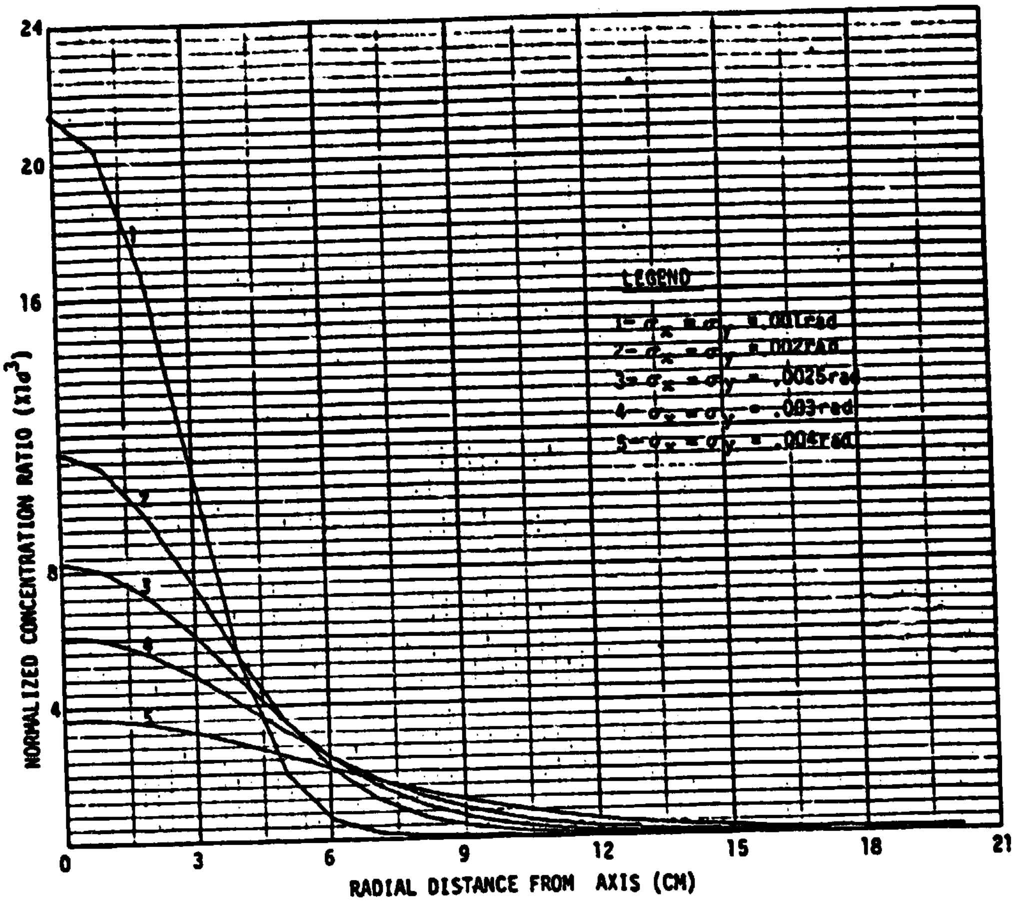


FIGURE (1) *Distribution of normalized concentration ratio along the focal plane using a two-dimensional normal distribution of slope errors ($\sigma_x = \sigma_y$).*

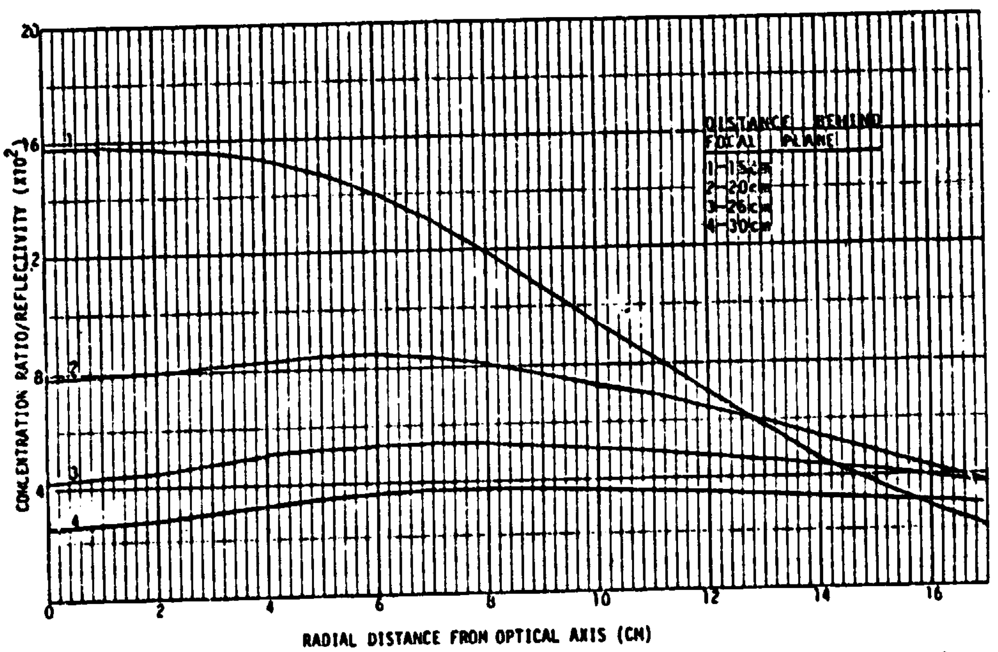
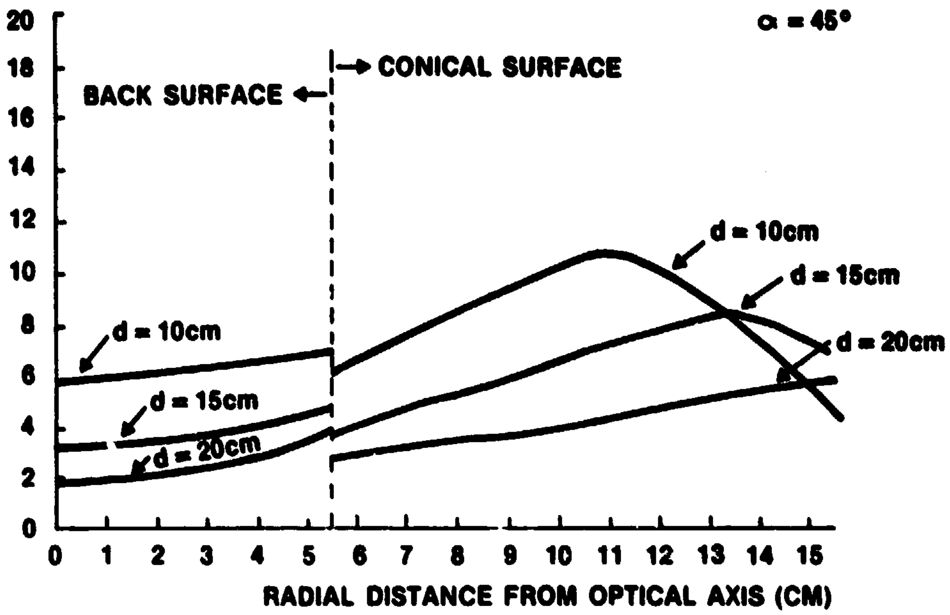
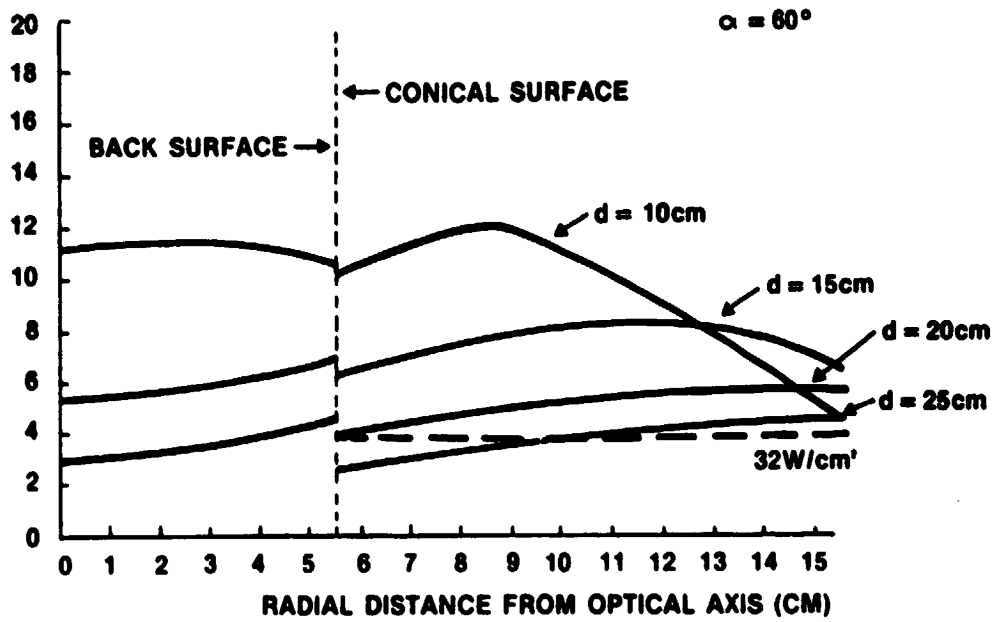


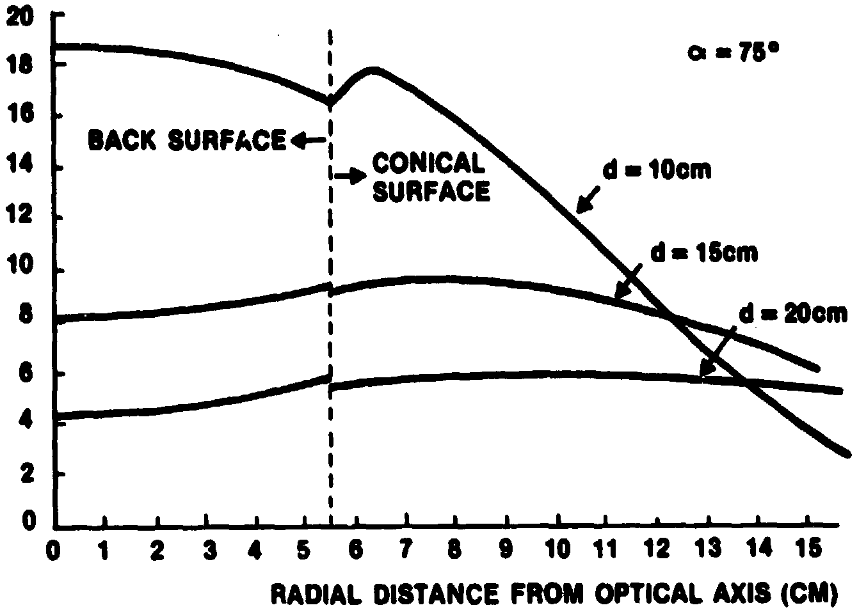
FIGURE (10) *Distribution of normalized concentration ratio along various planes located behind the focal plane as a function of radial distance from the optical axis ($\sigma_x = \sigma_y = .002$ rad).*

Special heat-flux distribution calculations for a P40 engine heater

P Poon, S Higgins, JPL.


α = 45°
CONICAL SURFACE
BACK SURFACE
d = 10cm
d = 15cm
d = 20cm
d = 10cm
d = 15cm
d = 20cm
0 2 4 6 8 10 12 14 16 18 20
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15
RADIAL DISTANCE FROM OPTICAL AXIS (CM)


α = 60°
CONICAL SURFACE
BACK SURFACE
d = 10cm
d = 15cm
d = 20cm
d = 25cm
32W/cm²
0 2 4 6 8 10 12 14 16 18 20
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15
RADIAL DISTANCE FROM OPTICAL AXIS (CM)


20
18
16
14
12
10
8
6
4
2
0
α = 75°
BACK SURFACE
CONICAL SURFACE
d = 10cm
d = 15cm
d = 20cm
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15
RADIAL DISTANCE FROM OPTICAL AXIS (CM)

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